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U. S. BATTLESHIP *RHODE ISLAND.*

DESCRIPTIONS—OFFICIAL TRIAL.

BY CHAS. B. EDWARDS AND RALPH L. LOVELL,
ASSOCIATES.

The *Rhode Island* is a twin-screw armored battleship built by the Fore River Shipbuilding Company of Quincy, Mass. The contract was signed February 15th, 1901, the contract price being $3,405,000, of which $1,123,650 was for machinery. This price does not include the armor and armor bolts (exclusive of the protective deck), the ordnance and ordnance outfit and certain articles supplied by the Government, but does include the fitting and installing of the above.

The time allowed for completion was 36 months, or to February 15th, 1904. For various causes, this time was extended to December 27th, 1905.

The main engines are to develop 19,000 indicated horsepower when making 120 revolutions per minute.

The guaranteed speed of the ship is 19 knots per hour for four hours. For failure to attain this speed a penalty was to

be inflicted at the rate of $50,000 per quarter knot between 18½ and 19 knots, and $100,000 per quarter knot between 18 and 18½ knots. If the speed fell below 18 knots the Government has the option of accepting the vessel at a reduced price.

The complement is 772 men and 49 officers.

PRINCIPAL DIMENSIONS OF HULL.

Length between perpendiculars, feet	435
on L.W.L., feet	435
over all, feet and inches	441– 1
Beam extreme, feet and inches	76– 2½
at load-water line, feet and inches	76– 2½
Depth molded, main deck, feet and inches	41– 6
Draught, mean normal, feet and inches	23– 9
on trial, feet and inches	23– 9
Displacement at trial draught, tons	14,680
normal draught, tons	14,680
per inch at L.W.L., normal draught, tons	61
Area of immersed midship section, square feet	1,737
L.W.L. plane, normal draught, square feet	25,610
Center of gravity of L.W.L. plane aft of midship section, feet	2.61
buoyancy above base line, feet	12.80
aft of midship section, feet	.19
Transverse metacenter above center of buoyancy, feet	19.13
Longitudinal metacenter above center of buoyancy, feet	515
Coefficient of fineness, block	.663
midship section	.96
L.W.L.	.772
Cylindrical coefficients, longitudinal	.690
vertical	.858
Number of frames	108

The keel of the *Rhode Island* was laid on March 1st, 1902, she was launched May 17th, 1904, the dock trial was July 22d, 1905, and the official trial for standardization November 1st, 1905, and four-hour trial November 11th, 1905.

The hull is built of basic open-hearth steel, with frames spaced 4 feet apart except under the machinery spaces. The inner bottom extends from frame 13 to frame 95, the double-bottom compartments between frames 60 and 69 are arranged for reserve feed-water tanks. A cofferdam 36 inches wide extends the whole length of the ship on the protective deck, extending 3 feet above the berth deck, packed with cellulose

forward and aft of the midship armor. The superstructure which supports the bridge deck extends between the 12-inch turrets, but is not the full width of the ship. It is recessed amidships for the broadside 8-inch turrets, and is kept back from the side of the ship sufficiently to allow of a fore-and-aft train of these turrets. Over the forward end of the superstructure is the bridge, and above that the flying bridge. On the forward bridge are the chart house, emergency cabin, conning tower and torpedo-directing stations. Two 3-pdr., two 30-caliber automatic and two 30-caliber machine guns are on this bridge. On the flying bridge are the compasses, steering apparatus and searchlights. There is a bridge over the after end of the superstructure on which are two 3-pdr. and four 30-caliber automatic guns. On the superstructure are four 3-pdrs. forward and the same number aft. The following boats are carried on the bridge deck:

1 50-foot steam cutter;
1 36-foot steam cutter;
2 36-foot launches;
4 30-foot cutters;
2 20-foot dingheys;
1 16-foot dinghey.

They are handled by two 5-ton, electrically-operated boat-cranes forward, and one 15-ton and one 8½-ton crane aft.

The main deck extends from stem to stern, and is partially covered by the superstructure between frames 32 and 78. The two superposed turrets with two 12-inch and two 8-inch guns each, and the two broadside turrets with two 8-inch guns each, are on this deck.

Mounted in the superstructure are four 3-inch guns. The hawse pipes extend through the bow to this deck, and the anchor chains lead to the windlass house, which projects through this deck. Inside the superstructure, from forward aft, are the crew's galley, drying room, bakery, officers' galley, paymaster's office, executive officers' office, officers' bathrooms, officers' waterclosets and wireless telegraph station. The following boats are carried on davits on this deck:

Two 30-foot cutters;

Two 20-foot dingheys forward and two 30-foot whaleboats;

One 30-foot barge and one 30-foot gig aft.

On the gun deck are mounted twelve 6-inch guns, four 3-inch guns forward and four 3-inch guns aft. Forward are the crew's shower baths and lavatories, amidships the berthing spaces, and aft the quarters of the admiral, captain, chief of staff, executive, navigator, chief engineer, paymaster, and two wardrooms. On the berth deck are forward, the sick bay, operating room, isolation ward, dispensary and bath, chief machinist's and chief petty officers' quarters and lavatories, staterooms of the master-at-arms and sergeant of marines, laundry, refrigerating plant and brig. Amidships is the crew space and aft the officers' quarters. The protective deck, which extends the full length of the ship, is 3 inches thick on the slope and 1½ inches on the flat. On the upper and lower platforms and in the hold are the magazines, storerooms, torpedo and mine rooms, etc. The protection of the vessel consists of an armor belt at the water line 8 feet in width, and a maximum thickness opposite the machinery spaces of 11 inches from the top down 5 feet, where it tapers to 8 inches at the bottom. Forward and aft of this belt the armor decreases in thickness by steps until at the bow and stern it has a uniform thickness of 4 inches. Above this main belt and covering the central portion of the ship for a distance of 245 feet is armor of a uniform thickness of 6 inches extending up to the main deck and connected to the aft barbette by athwartship armor 6 inches thick, and joined forward by inclined armor 6 inches thick, thus forming a central casemate.

The 12-inch barbettes, which are supported on the protective deck, are 10 inches thick in front and 7½-inches thick in back. The 8-inch barbettes, which rest on the main deck, are 6 inches thick in front and 4 inches in back. The subbarbette, from the main to the gun deck, is 4 inches thick. From the gun deck an armored tube protecting the ammunition hoists extends through the protective deck.

The 12-inch guns are protected by armor 12 inches thick on the slope, 8 inches thick on the side and 6 inches thick

at the back. The armor for the 8-inch superposed guns is 12-inches thick on the slope, 8 inches on the sides and 6 inches in back. The armor for the 8-inch broadside guns is 6½ inches thick on the slope, 6 inches thick on the sides and back.

ELECTRIC PLANT.

The generating plant consists of two dynamos of 100 kilowatt and six of 50 kilowatt capacity at 125 volts. They are in four rooms, two at the level of the hold and two above, with the center-line bulkhead separating each pair.

The generators are of General Electric make, while the engines are from B. F. Sturtevant Co.

The lighting outfit comprises in the various circuit fixtures for 1,154 lights and 6 searchlights.

The power outfit comprises 112 motors, as follows:

4 turret-turning motors of 35 H.P. each;
4 turret-turning motors of 15 H.P. each;
4 ammunition-hoist motors of 30 H.P. each;
8 ammunition-hoist motors of 8 H.P. each;
21 ammunition-hoist motors of 2½ H.P. each;
5 whip-hoist motors of 3 H.P. each;
4 gun-elevating motors of 5 H.P. each;
8 gun-elevating motors of 3½ H.P. each;
4 rammer motors of 5 H.P. each.
8 rammer motors of 5 H.P. each;
5 boat-crane motors of 35 H.P. each;
3 boat-crane motors of 20 H.P. each;
1 workshop motor of 10 H.P.
33 ventilation-fan motors varying in sizes from ½ H.P. to 15 H.P. each.

DYNAMO ENGINES.

Type	Compound.
100 kw. dynamos, number	2
Diameter of cylinder, H.P., inches	10
L.P., inches	18
Stroke, inches	10
50 kw. dynamos, number	6
Diameter of cylinder, H.P., inches	8
L.P., inches	14
Stroke, inches	8

PROPELLING MACHINERY.

There are two four-cylinder triple-expansion outboard-turning engines of the vertical, inverted, direct-acting type, in two watertight compartments, separated by a fore-and-aft watertight compartment. The order of the cylinders from forward is F.L.P., H.P., M.P., A.L.P. The H.P. and M.P. cranks are at 90 degrees, as are also the two L.P. cranks. The H.P. and F.L.P. cranks are opposite. The sequence of the cranks is H.P., M.P., F.L.P., A.L.P. The cylinders are supported by twelve forged-steel columns, which are tied fore and aft by heavy cast-steel I beams on the back supporting the crosshead guides, and on the front by a forged-steel tie. Additional forged-steel diagonal ties are fitted on both sides. Each pair of columns is tied together athwartship by diagonal braces. The valves are of the single-ported piston type, with cast-iron heads and packing rings and steel followers. The H.P. valve takes steam on the inside, the others on the outside. The H.P. cylinder has one valve, the others two each. All valves are arranged to draw from the top. Each valve has a balance piston working in a cylinder cast in the cover. The valve gear is Stephenson double-bar link type, and the H.P. valve stem connects with the link direct, while the M.P. and L.P. valve stems are connected by crossheads.

The engine is controlled by a floating-lever reversing engine with oil cylinder control, and is connected to a reverse shaft supported from the engine columns. The reverse shaft carries arms in which independent adjustable cutoff blocks are fitted to connect with each link, giving a range in cutoff from .5 to .74 of the stroke. All the cylinders are jacketed around the working liners and at both ends, except the H.P., which has no jacket on the ends.

Each engine has a throttle valve balanced by a piston working in a cylinder cast with the valve casing, and operated by a hand wheel at the working platform. The H.P. piston is of cast iron, the others of cast steel, all having forged-steel followers. Each piston has two packing rings, each in one piece cut diagonally, with tongue pieces over the joint. The rings

are put in the pistons so that the joints come opposite each other. The piston rods are hollow, of class A nickel-steel, oil tempered and annealed. They are ground cylindrical and polished and tapered at each end to fit the piston and the crossheads. The crossheads are of nickel-steel forged with the crosshead pins and are bolted to cast-steel slippers. The go-ahead guides are of cast iron, hollow, to provide for water circulation, and are supported at the top by lugs on the cylinders and at the bottom by the cast-steel tie rods. The backing guides of cast iron are bolted to flanges on the go-ahead guides. The connecting rods are of the same material as the piston rods, are hollow and forked at the upper end to carry the crosshead brasses. The crank-pin boxes are of composition and are held to the stub end of the connecting rod by a steel cap and through bolts. The eccentrics are in two parts, the larger of cast iron and the smaller of wrought steel. The eccentric straps are of composition and the rods of nickel-steel.

The turning engine is a double, inverted engine located at the aft end of the bedplate and connected to the crankshaft by worm gearing. The working platforms are inboard, where are located the throttle-valve wheel, reversing and drain levers, clock, gauges, counters, etc.

ENGINE DATA.

Cylinders, number for each engine	4
H.P. diameter, inches	35
M.P. diameter, inches	57
L.P. diameter inches	66
Stroke of all pistons, inches	48
Valves, H.P., one for each cylinder, inches	20
M.P., two for each cylinder, inches	22
L.P., two for each cylinder, inches	24
Valve stems, H.P. diameter, inches	3
M.P. diameter, inches	3
L.P. diameter, inches	3
diameter through valve, H.P. and I.P., inches	$1\frac{1}{16}$
L.P., inches	$2\frac{1}{4}$
at balance piston, H.P. and I.P., inches	$1\frac{1}{4}$
L.P., inches	$1\frac{3}{8}$
Main steam pipe, diameter at throttle, inches	12
Area of cross section, square inches	113

Net area of both M.P. to H.P. piston	2.65
both L.P. to H.P. pistons	7.11
Piston-rod diameter, inches	8
of axial holes, H.P., inches	1¼
M.P., inches	3⅜
L.P., inches	5
length, feet and inches	9- 4 5/16
Cylinder wall, H.P., thickness, inches	2
M.P., thickness, inches	1½
L.P., thickness, inches	1½
liners, H.P., thickness, inches	1½
M.P., thickness, inches	1⅝
L.P., thickness, inches	1¼
Valve-chest liners, thickness, inches	1½
Jacket space, inch	¾
Cylinder relief valves, two to each cylinder, inches	3½
Connecting rod, length center to center, inches	96
diameter upper end, inches	7¾
lower end, inches	9
axial hole, H.P., inches	1½
M.P., inches	4¼
L.P., inches	5
crosshead bolts (four), inches	3¼x26½
crank pin bolts (two), inches	4¼x37½
Crosshead surface, each, square inches	690
backing, each, square inches	525
pin diameter, inches	9¼
diameter of axial hole, H.P., straight, inches	1½
I.P. and L.P., double taper; inches	2 to 6
length of pin, inches	10 1/16
Crankshaft, number of sections	2
diameter, inches	17½
of axial hole, inches	10½
coupling dics, inches	32
thickness coupling discs, inches	3⅝
coupling bolts in one flange (eight), diameter, inches	3½
journals, diameter, inches	17½
length forward to aft, inches, 21½, 21½, 21½, 21½, 29, 21½	
length of forward section, feet and inches	16- 6¼
after section, feet and inches	17-10
Crank pin, diameter, inches	18
length, inches	21½
axial hole, inches	10½
webs, width, inches	20
thickness, inches	11½
Thrust shaft, diameter, inches	16¼
axial hole, inches	10½

Thrust shaft, collars, number, on each shaft	13
diameter, inches	26
thickness, inches	2
space between, inches	4
total thrust surface each engine, sq. inches	3,952
length, feet and inches	14- 9
steady bearing, diameter, inches	16½
length, inches	12
Line shaft, diameter, inches	16½
axial hole, inches	10½
coupling disc, inches	32
coupling bolts in one flange	8
diameter, inches	3½
length, feet and inches	29- 0
Propeller shaft, without casing, diameter, inches	17½
in bearings, inches	19½
length, feet and inches	26- 3
Stern-tube shaft, after bearing, diameter, inches	19½
length, feet	6
forward bearing, diameter, inches	19½
length, feet and inches	5- 7
Inboard coupling, diameter, inches	32
length, inches	19½
Outboard coupling, diameter, inches	23
length, feet and inches	6- 0
Stern-bracket bearing, diameter, inches	19½
length, feet and inches	7- 2
Reversing engine, diameter steam cylinder, inches	15
oil cylinder, inches	7½
stroke, inches	21½
Turning engine, diameter cylinders (two), inches	6
stroke, inches	6
Clearance of H.P. cylinder, per cent., top, 24.7; bottom,	23.7
M.P. cylinder, per cent., top, 21.; bottom,	20.7
F.L.P. cylinder, per cent., top, 19.4; bottom,	29.2
A.L.P. cylinder, per cent., top, 19.3; bottom,	19.4

MAIN CONDENSERS.

There is one main condenser in each engine room. The shell is of steel, the water heads of composition, the tubes of Admiralty metal not tinned. There is a by-pass valve in the division plate of the water chest at the injection and delivery end to allow the circulating pump to discharge directly overboard.

Diameter of shell, inside, feet and inches	6–10
Thickness of shell, inch	$\frac{8}{8}$
Length over heads, feet and inches	15– 8
Tubes, diameter, outside, inch	$\frac{5}{8}$
length between tube sheets, feet and inches	13– 0
thickness, inch	1
spacing between centers, inch	$1\frac{1}{16}$
number in each condenser	5,583
cooling surface, each, square feet	11,875

MAIN AIR PUMPS.

Forward of the condenser in each engine room there is a two-cylinder, vertical, double-acting air pump of the Fore River design.

Diameter of steam cylinders, each, inches	14
water cylinders, each, inches	$31\frac{1}{2}$
Stroke, inches	21
Diameter of piston rods, inches	$2\frac{1}{2}$
pump rods	$4\frac{1}{2}$
pump valves in bucket	$5\frac{1}{2}$
Number of delivery valves, 30; diameter of each, inches	$5\frac{1}{2}$
Area through valve openings, square inches	16

MAIN CIRCULATING PUMPS.

There is one centrifugal, double-inlet, circulating pump driven by compound, vertical, inverted engine, located aft of the main condenser. Each pump has a capacity of 1,200 gallons per minute, at 266 revolutions per minute, and it may draw from the sea, main drain or the engine-room bilge. The three suctions are so interlocked that only one can be open at a time.

Diameter of H.P. cylinder, inches	10
L.P. cylinder, inches	20
Stroke, inches	10
Diameter of pump runner, inches	42
Width of pump runner at periphery, inches	5
hub, inches	12
Diameter of inlet nozzle, inches	20
outlet nozzle, inches	$19\frac{1}{2}$

FEED AND FILTER TANKS.

Under each condenser is a combined feed and filter tank of 4,200 gallons capacity. A horizontal partition divides each tank into two compartments, of which the upper is the filter

tank, of 1,150 gallons capacity, and the lower the feed tank proper, of 3,050 gallons capacity. Each filter tank is divided into five parts by vertical partitions, and the water flows through the filtering material four times on its way to the tank.

FEED PUMPS.

There are two main feed pumps of Fore River design in each engine room. They draw from the hotwell-pump discharge and discharge into the main feed lines.

In each of the six fire rooms is one auxiliary feed pump of Fore River design bolted to the center-line bulkhead. Each pump may draw from the feed tanks, from the hotwell-pump discharge, from the main drain, the auxiliary drain, the sea, the bilge in its own compartment and from the bottom blows of all boilers in its compartment. It may discharge into the auxiliary feed line, the fire main or overboard.

Main feed pumps, double-acting, single, vertical, piston type.

Diameter of steam cylinder, inches	18
water cylinder, inches	10
Stroke, inches	18

Auxiliary feed pumps, double-acting, single, vertical, piston type.

Diameter of steam cylinders, inches	14
water cylinders, inches	8
Stroke, inches	12

FEED-WATER HEATERS.

In each engine room there is one feed-water heater containing 950 square feet of heating surface in 666 tubes. Steam from the auxiliary exhaust is by-passed into the heater and the back pressure is regulated by a spring relief valve. The hotwell pump delivers water through this heater to the feed-pump suction.

HOTWELL PUMP.

There is one vertical, double-acting, single-piston, hotwell pump in each engine room, which may draw from either air-pump suction, from hose connections on the side of the ship, and from the main and reserve-feed tanks. It may discharge

into the feed-pump suction either directly or through the feed-water heater, into the auxiliary feed line or into the main or reserve feed tanks.

Diameter steam cylinder, inches	12
water cylinder, inches	14
Stroke, inches	12

FIRE AND BILGE PUMPS.

There is one vertical, double-acting, single-piston, packed fire and bilge pump in each engine room. These pumps draw from the sea, the drainage system and from the bilge. They discharge into the fire main or overboard.

Diameter of steam cylinder, inches	12
water cylinder, inches	10
Stroke, inches	12

AUXILIARY CONDENSERS.

In each engine room there is one auxiliary condenser having 602 square feet of cooling surface, with combined air and circulating pump attached.

Diameter of steam cylinder, inches	8
water cylinder, inches	10
air cylinder, inches	10
Stroke, inches	12

In the dynamo room there is one auxiliary condenser having 1,207 square feet of cooling surface, with a combined air and circulating pump attached, of the following dimensions:

Diameter of steam cylinder, inches	12
water cylinder, inches	14
air cylinder, inches	14
Stroke, inches	12

FORCED DRAFT BLOWERS.

The closed fire-room system of forced draft is used, there being two Sturtevant blowers for each fireroom.

Number of blowers	12
Type of engine	Two-cylinder, vertical.
Diameter of steam cylinders, inches	6
Stroke, inches	5
Diameter of piston rods, inches	$1\frac{5}{16}$
fan, inches	66

MACHINE SHOP.

Located on the upper platform between the aft fire room and the engine room and extending athwartship to the width of the engine room is the general workshop. This is divided into two parts by a fore-and-aft watertight bulkhead. Communication between the two is by means of a watertight door. The tools are driven by a 10-horsepower motor. The following tools are installed:

1 30-inch and 59-inch swing back-geared gap lathe, 10 feet between centers;
1 14-inch screw-cutting back-geared engine lathe, 4 feet between centers;
1 16-inch back-geared crank shaper;
1 36-inch Dreses simplex radial drill, to drill up to 1½ inches;
1 16-inch sensitive drill;
1 No. 1 Universal milling machine;
1 combined hand punch and shears;
1 double emery grinder;
1 30-inch grindstone;
7 bench vises.

BLACKSMITH SHOP.

Located on upper deck amidships, one motor-driven forge and the usual equipment of tools is installed.

ASH HOISTS.

There are six ash hoists, made by Williamson Brothers. They are located in the fireroom hatches, and the buckets are hoisted through the ventilators to the gun deck and trollied to the ash chutes at the sides.

EVAPORATING AND DISTILLING PLANT.

There are four evaporators of the Williamson Brothers make, each having a capacity of 4,750 gallons in 24 hours and having a heating surface of 240 square feet each. There are six distillers of a combined capacity of 20,000 gallons of water in 24 hours, each having a cooling surface of 66 square feet.

The evaporators are located on the berth deck at the after end of the boiler hatch, and the distillers are located in the hatch one deck higher.

ICE MACHINE.

One Allen dense-air ice machine of three tons daily capacity is installed forward on the port side of the berth deck, with the ice-making boxes near. Cooling pipes lead to the four refrigerating rooms and to the scuttle butts.

SCREW PROPELLERS.

The propellers are of manganese-bronze, and turn outboard when going ahead, the starboard propeller being right-hand. They are true screws with the surfaces ground smooth and tinned.

Number of blades	3
Diameter, feet and inches	17- 9
of hub, feet and inches	4- 8
Length of hub, feet and inches	3- 7
Pitch, feet and inches	18- 8
Helicoidal area, square feet	99.2
Projected area, square feet	82.5
Pitch ratio	1.06
Area ratio	.4

BOILERS.

There are twelve Babcock & Wilcox boilers of the latest improved type, arranged in six watertight compartments.

Floor space, length, feet and inches	9- 1½
width, feet and inches	17- 9½
Length of grate, feet and inches	7- 0
With of grate, feet and inches	16- 0
Grate surface one boiler, square feet	111.8
all boilers, square feet	1,342
Heating surface, one boiler, square feet	4,682
all boilers, square feet	56,184
Ratio H.S. to G.S.	41.87
Number of tubes, one boiler, 2-inch	946
4-inch	62
total	1008
elements one boiler	30
Weight of water in one boiler at steaming level, pounds	16,454
one boiler complete with fittings, pounds	96,545

HEATING SYSTEM.

Radiators and heaters throughout the ship are arranged in circuits, each circuit being so connected that it can be operated independently of the others. The circuits are in turn arranged in groups, each group having a receiving manifold and a drainage manifold. Steam is taken from the auxiliary steam pipe at the receiving manifold. Here there is a stop valve for each circuit and a stop valve, an adjustable reducing valve, and a steam gauge at the connection with the auxiliary steam pipe. The drainage manifolds have a stop valve and a check valve for each circuit. Each manifold discharges through a trap with a by-pass to the feed tanks, the auxiliary condenser or the bilge. By this arrangement each circuit can be entirely isolated and water from other circuits prevented from backing up into it.

OFFICIAL TRIALS.

Standardization of Screws.—The vessel was tried by the standardized-screw method, progressive runs being made over the measured mile at Rockland, Maine, November 2, 1905. The data recorded in Tables 1 and 2 were obtained. From these the curves on Plate I (furnished by the Bureau of Steam Engineering) and Plate II were plotted.

It was determined that 125.7 revolutions per minute of main engines would be required to give the contract speed of 19 knots.

The draught and corresponding displacement at the beginning and end of runs were as follows :

	Beginning.	End.
Draught, forward, feet and inches	23- 9¼	23- 4¼
aft, feet and inches	23- 9¼	23- 8¼
Displacement, tons	14,955	14,784

The coal used was Pocahontas, run of the mine.

OFFICIAL FOUR-HOURS' TRIAL.

On November 3, 1905, the *Rhode Island* got underway at 8·30 A. M., and started out to sea for the four-hours' trial. At 10·27 A. M. the trial commenced, but at the end of the

three hours was discontinued owing to leaky condensers. The vessel then proceeded to an anchorage off Boston Light, where she remained until November 11, 1905, when, being reported ready for trial, she put to sea. At 10·32 A. M. the trial commenced, finishing at 12·32 P. M., completing the four hours required. Average revolutions per minute during the trial was 125.86, corresponding to a speed of 19.011 knots.

The draught and displacement at the beginning of the trial were:

Draught, forward, feet and inches	23- 6¼
aft, feet and inches	24- 1¼
mean, feet and inches	23-10¼
Corresponding displacement, tons	15,015

The estimated displacement at the end of the trial was 14,600 tons.

A synopsis of the data obtained follows.

PERFORMANCE.—FOUR-HOURS' OFFICIAL TRIAL.

Steam pressures. (*Average one-half hourly observations*).

	Starboard.	Port.
Steam pressure at boilers (per gauge), pounds	268.6	
H.P. steam (per gauge), pounds	288.8	244.3
1st receiver (absolute), pounds	113.2	114
2d receiver (absolute), pounds	40.7	41.3
Vacuum in condensers, inches of mercury	26.5	25.56

Mean effective pressures in cylinders, in pounds per square inch.

Main engine:
Starboard.....H.P., 88.81; I.P., 50.08; F.L.P., 18.45; A.L.P., 13.76
Port.....H.P., 94.1; I.P., 50.74; F.L.P., 18.34; A.L.P., 15.9
Mean pressure, in pounds per square inch on L.P. piston, equivalent to aggregate M.E.P. on all pistons, starboard..... 47.02
Mean pressure, in pounds per square inch on L.P. piston, equivalent to aggregate M.E.P. on all pistons, port..... 49.03

Revolutions per minute.

Main engines.....starboard, 125.291; port, 126.437
Pumps, air.....starboard, 23.7; port, 23.4
circulating.....starboard, 182.0; port, 211.1

Pumps, feed, double strokes per minute	starboard, inboard,	32.55
	outboard,	32.55
	port, inboard	21.1
	outboard	38.0
Blower engines, average of twelve blowers		492.65
Speed of ship, in knots per hour		19.01
Slip of propeller, in per cent. of its own speed, based on mean pitch,	starboard, 16.88; port,	17.64
Air pressure in firerooms, in inches of water		1.9

Indicated Horsepower.

Main engines:		
Starboard	H.P., 2,526.9; I.P., 3,843.5; F.L.P., 1,903.5; A.L.P.,	1,419.5
Total		9,693.3
Port	H.P., 2,702.2; I.P., 3,930.9; F.L.P., 1,908.6; A.L.P.,	1,654.6
Total		10,196.3
Grand total		19,889.6
Air pump	starboard, 7.8; port,	7.75
Circulating pump	starboard, 54; port,	81
Feed pumps	starboard, 140; port,	130
Dynamo condenser, combined air and circulating pump		13.5
Hotwell pumps	starboard, 9.3; port,	6.6
Distiller circulating pump		6.4
Fire and bilge pump	port,	4.5
Blower engines, for forced draft in firerooms, 12 blowers		198
Dynamo engines		78.5
Collective of all main engines		19,889.6
main engines, air, circulating and feed pumps		20,310.15
and auxiliary engines in operation on trial		20,626.95
all machinery during trial, per square foot of	G.S.	15.37
	H.S.	.36

Coal.

Kind and quality	Pocahontas, run of the mine.

The working of the machinery, both main and auxiliary, and its performance during the trial, were entirely satisfactory. The boilers steamed finely and worked in a satisfactory manner throughout.

A careful inspection of the various parts of the machinery during and after the trial showed the engines, boilers and appurtenances to be in excellent condition.

The leaking of the condensers during the unsuccessful trial having been found to be due to having no scattering plates, this omission was rectified, and the working of the condensers found to be entirely satisfactory.

Table 1.—U. S. S. *RHODE ISLAND.*

Data from Progressive Runs over Measured-Mile Course off Rockland, Maine, November 2, 1905.

Number and direction of run.	Time on course.	Revolutions of main engines.					Speed in knots per hour.	Indicated horsepower.											
		To make 1 knot.		Per minute.				Starboard engine.					Port engine.						
		Starboard.	Port.	Starboard.	Port.	Mean.		H.P. cylinder.	I.P. cylinder.	Forward L.P. cylinder.	After L.P. cylinder.	Total.	H.P. cylinder.	I.P. cylinder.	Forward L.P. cylinder.	After L.P. cylinder.	Total.	Total of both main engines.	Total all auxiliaries.	Total all machinery.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21
	m. s.																			
1-S.	3-16.2	392.0	396.55	119.89	121.27	120.58	18.349	3,131.9	2,724.4	1,421.4	1,421.4	8,699.1	2,975.0	2,934.1	1,447.7	1,347.9	8,704.7	17,403.8	560.4	17,964.2
2-N.	3-10.7	388.65	395.15	122.31	124.35	123.33	18.882	3,361.5	2,996.5	1,550.6	1,575.8	9,484.4	3,065.0	3,123.0	1,638.2	1,638.2	9,464.4	18,948.8	613.9	19,562.7
3-S.	3-14.2	395.10	400.30	121.91	123 52	122.72	18.542	3,129.2	3,173.6	1,736.4	1,726.4	9,765.6	2,974.2	3,026.4	1,627.1	1,678.0	9,305.7	19,071.3	620.7	19,692.0
4-N.	3-06.3	384.60	391.00	124 00	126.03	125.02	19.329	2,929.3	3,683.7	1,980.5	1,898.8	10,492.3	2,719.8	3,906.2	1,867.7	2,075.3	10,569.0	21,061.3	609.4	21,670.7
5-S.	3-14.8	400.95	402.95	123.56	124.17	123.87	18.483	2,750.6	3,632.8	1,953.2	1,922.6	10,259.2	2,708.0	3,470.6	1,830.0	1,830.0	9,838.6	20,097.8	660.5	20,758.3
6-N.	3-06.8	386.90	391.90	124.54	126.15	125.34	19.272	2,885 5	3,776.1	2,153.2	2,025.0	10,839.8	2,607.7	3,709.0	2,077.2	2,284.9	10,678.8	21,518.6	659.0	22,177.6
7-S.	...	...	...	...	...	...	...	3,340.8	2,943.5	1,668.5	1,638.5	9,591.3	3,145.2	2,792.0	1,513.2	1,624.2	9,074.6	18,665.5	657.1	19,322.6
8-N.	3-11.8	...	...	...	...	...	18.770	3,153.3	2,920.6	1,560.2	1,560.2	9,194.5	3,035.0	2,715.5	1,330.0	1,270.0	8,350.5	17,545.0	588.5	18,133 5
9-S.	3-21.6	...	...	...	...	...	17.857	2,669.8	2,509.0	1,370.0	1,370.0	7,918.8	2,682.5	2,363.3	1,184.0	1,145.5	7,375.3	15,294.1	563.1	15,857.2
10-N.	3-14.4	378.60	380.30	116.85	117.38	117.12	18.518	2,880.0	2,719.7	1,407.5	1,462.3	8,469.5	2,776.1	2,381.8	1,127.7	1,215.2	7,500.8	15,970 3	506.0	16,476.3
11-S.	3-25.4	383.80	393.75	112.10	115.02	113.56	17.527	2,482.7	2,204.2	1,135.2	1,144 4	6,966.5	2,534.3	2,254.5	1,022.7	1,202.7	7,014.2	13,981.7	441.5	14,423.2
12-N.	3-27.9	376.90	374.55	108.81	108.13	108.47	17.320	2,175.0	2,066.1	949.6	994.37	6,185.07	2,014.0	1,589.4	765.5	694 3	5,063.2	11,248.27	479.4	11,727.67
13-S.	3-39.9	378.60	380.10	103.30	103.71	103 51	16.371	1,865.5	1.531.2	782.44	901.5	5,080.64	1,837.5	1,397.5	725.8	853.8	4,814.6	9,895.24	459 1	10,354.34

13-S.		3-39.9	378.60	380.10	103.30	103.71	103 51	16.371	1,865.5	1.531.2		782.44	901.5	5,080.64	1,837.5	1,397.5	725.8	853.8	4,814.6	9,895.24	459 1	10,354.34													

Table II.—U. S. S. *RHODE ISLAND.*

Data from Progressive Runs over Measured-Mile Course off Rockland, Maine, November 2, 1905.

Number and direction of run.	Which engine.	Steam pressure.					Mean effective pressure.					Vacuum in inches.	Air pressure in inches of water.	Cut-off in cylinders from beginning.				Temperatures.							Revolutions or double strokes.											
		In boilers above atmosphere.	At engines above atmosphere.	H.P. steam chest, absolute.	In 1st receiver, absolute.	In 2d receiver, absolute.	H P. cylinder.	L.P. cylinder.	Forward L.P. cylinder.	After L.P. cylinder.	Referred to L.P.			H.P.	I.P.	Forward L.P.	After L P.	Outside air.	Engine-room working platform.	Fireroom working level.	Injection.	Discharge.	Hotwell.	Feed.	Dynamo.	Circulating pump.	Main feed pump.	Auxiliary feed pump.	Hotwell pump.	Water-service pump.	Bilge pump.	Air pump.	Auxiliary condenser pump.	Lynamo condenser pump.	Fireroom blowers.
35	36	37	38	39	40	41	42	43	44	45	46	47	48	49	50	51	52	53	54	55	56	57	58	59	60	61	62	63	64	65	66	67	68	69	70
1-S.	S.	257	250	250	87	35	115	37.1	14.4	14.4	44 06	25.5	1 23	.74	.74	.65	.64	38	66	60 5	49	110	90	154	350	130	24	25	34	25	23	22	30	25	440
	P.	257	250	249	92	35	108	39.5	14.5	13 5	43.59	26 5		.74	.74	.65	.65		63		50	94	80	186		206	30		54			64			385.8
2-N.	S.	269	256	258	92	35	121	40	15.4	15.65	47 10	25.5	1.39	.74	.74	.65	.64	38	64	58.5	49	109	88	148	350	146	26	26	37	24	23	21	44	25	431.6
	P.	269	265	265	97	38	108.5	41	16	16	46.22	26.5		.74	.74	.65	.65		66		50	98	84	188		190	31		40			44			475 8
3-S.	S.	263	256	257	95	37	113	42.5	17.3	17.2	48.65	25 5	1.46	.74	.74	.65	.64	38	69	58.8	49	109	86	144	350	165	32	32	41	22	12	21	39	25	411.6
	P.	263	269	270	102	40	106	40	16	16.5	45.75	26		.74	.74	.65	.65		66		50	100	82	190		190	42		48			34			452.5
4-N.	S.	265	256	259	112	43	104	48.5	19 4	18.6	51 39	25	1.54	.74	.74	.65	.64	38	64	58.3	49	109	84	143	350	160	32	32	40	24	18	20	48	44	420
	P.	265	269	270	112	43	95	50.6	18	20	50.93	26		.74	.74	.65	.65		66		50	100	82	188		196	40		43			48			435.8
5-S.	S.	261	243	244	107	40	98	48	19.2	18.9	50.43	25	1.57	.74	.74	.65	.64	38.5	70	59.5	49	109	86	144	350	156	33	30	40	24	26	20	42	35	445
	P.	261	250	249	107	40	96	44.8	17.9	17.9	47.81	26		.74	.74	.65	.65		68		50	102	82	188		200	46		45			42			461.6
6-N.	S.	265	256	265	114	42	102	49.5	21	19.75	52.86	25	1 77	.74	.74	.65	.64	39	70	59.5	49	110	102	170	350	150	32	31	40	24	25	26	39	25	4[illegible]3
	P.	265	265	265	117	45	91	44	20	22	51.41	26		.74	.74	.65	.65		70		50	104	82	...		190	54		54			52			435
7-S.	S.	268	264	265	94	38	121.2	39.6	16.7	16.4	48	22	1.61	.74	.74	.65	.64	38	72	60.8	49	108	94	158	350	162	32	30	40	24	22	24	58	33	441.6
	P.	268	265	265	92	35	113	37.2	15	16.1	44.98	26.5		.74	.74	.65	.65		68		50	102	84	...		192	52		52			58			451.6
8-N.	S.	261	245	249	97	35	115	39.5	15.7	15.7	46.25	25.5	1.58	.74	.74	.65	.64	38	72	58.7	49	100	88	140	350	150	26	22	37	15	21	25	33	25	463.3
	P.	261	250	249	87	35	110	36.5	13.3	12.7	41.75	26.5		.74	.74	.65	.65		70		50	100	85	190		186	29		33			54			436.6
9-S.	S.	227	228	224	81.5	30	101	35.2	14.3	14 3	41.32	25.5	1.32	.74	.74	.65	.64	39	72	58.8	43	104	88	140	350	142	37	34	40	13	13	23	33	25	415
	P.	227	226	224	82	30	101	33	12.3	11.9	38.31	26.5		.74	.74	.65	.65		70		50	100	85	...		170	28		32			40			441.6
10-N.	S.	235	230	232	81.5	33	108.5	38	14.63	15.2	44.03	25	1.44	.74	.74	.65	.64	38	73	60.5	49	103	82	132	350	142	36	34	48	13	14	22	18	25	433.3
	P.	235	235	234	82	30	103.5	33	11.6	12.5	38.60	26		.74	.74	.65	.65		70		50	106	83	...		150	12		15			44			416 6
11-S.	S.	238	228	219	69.8	30	97 5	32.1	12.3	12 4	37.74	21	1.17	.74	.74	.65	.64	40	76	59 5	49	104	88	140	350	134	31	31	55	14	12	29	15	25	233.3
	P.	238	240	224	77	30	97	32	10.8	12.7	37.03	26.5		.74	.74	.65	.65		68		50	104	84	...		168	19		35			50			281.6
12-N.	S.	193	189	189	65.5	25	88	31	10.6	11.1	34.52	26	1.06	.74	.74	.65	.64	41	78	61.8	49	103	90	110	350	140	27	30	33	12	26	24	16	25	366.6
	P.	193	184	183	66	25	82	24	8.6	7.8	28.44	26		.74	.74	.65	.65		68		50	94	85	...		160	20		27			30			423.3
13-S.	S.	219	215	172	65.5	25	79.5	24.2	9.2	10.6	39.86	26	.89	.74	.74	.65	.64	40	78	61.5	49	100	94	136	350	146	22	24	43	13	20	26	17	25	295
	P.	219	209	173	56	22	78	22	8.5	10	28.19	26.5		.74	.74	.65	.65		68		50	90	84	...		174	26		33			43			381.6

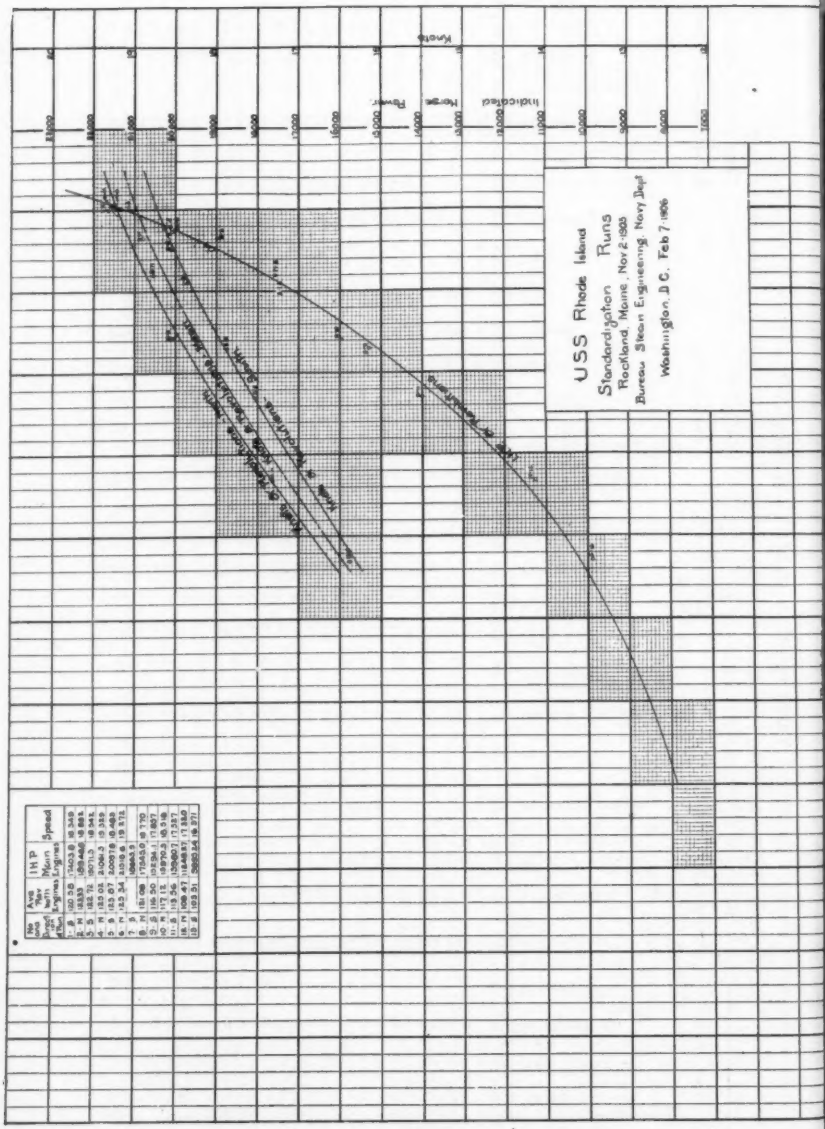
USS Rhode Island
Standardization Runs
Rockland, Maine, Nov 2-1905
Bureau Steam Engineering, Navy Dept
Washington, D.C. Feb 7 1906

Plate I.

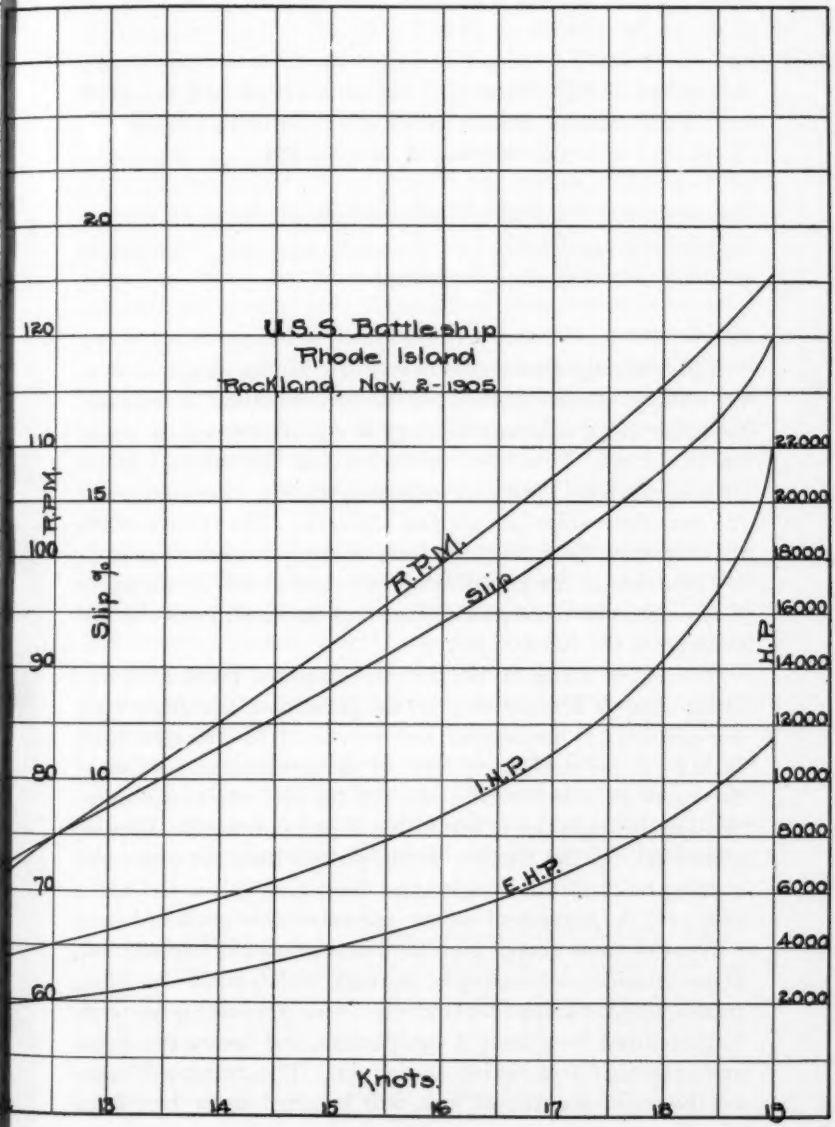
U.S.S. Battleship
Rhode Island
Rockland, Nov. 2-1905
R.P.M.
Slip %
H.P.
Knots.
R.P.M.
Slip
I.H.P.
E.H.P.
120
110
100
90
80
70
60
20
15
10
22000
20000
18000
16000
14000
12000
10000
8000
6000
4000
2000
13
14
15
16
17
18
19

Plate II.

AN INVESTIGATION OF THE PRESSURES UPON THE MAIN BEARINGS AND CRANK PINS OF MARINE ENGINES.

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This investigation was undertaken with the object of finding an approximate method for the determination of the loads to which the crankshaft bearings of a proposed engine would be subjected. Data were collected for the naval engines *Monterey*, *New York* and *Olympia*, and for the engines of the merchant ships *Boston* and *Siberia*. The results of the investigation show that the load on any crankshaft bearing is a function of the power being transmitted and developed at that point, and consequently increases as we go from the first cylinder to the last cylinder.

The forces acting to produce the resultant pressure at any instant upon a bearing are; (1) the pressure of the steam upon the piston; (2) the weight and inertia of the reciprocating parts; (3) the centrifugal force of the unbalanced portions of the crank webs and crank pin; (4) the load due to the transmission, through the crank webs, of power from the forward cylinders; (5) the friction of the piston rings and piston-rod stuffing box; (6) the weight and inertia of valves and valve gear; (7) the friction of valves and valve-stem stuffing boxes.

None of these except possibly No. 4 needs any explanation. If we consider a crank pin through which power is being transmitted, we know that since the crank is revolving uniformly there must be a state of equilibrium, and hence two equal and opposite forces acting at the pin. The crank web nearest the resistance, the aft web, will be acted upon by a force

in the direction of motion. The crank web nearest the power, the forward web, will be acted upon by a force whose direction is opposite to the direction of motion. These two forces acting at the pin are also acting upon the adjacent bearings.

Only the first four of the forces mentioned above have been considered in working up all the engines, in order that the results obtained might be of general application. There is no very exact method for determining the loads due to friction, and the effect of the valve gear will vary depending upon whether steam is taken at the ends or at the middle. It therefore seemed best to treat only those forces about which there could be no question, and then to modify the results as seemed best from a consideration of these forces which are variable and capable of less exact treatment. A separate calculation was made in the case of the *Siberia's* engines to determine what effect the friction of the pistons and valves, and the weight and inertia of the valve gear, had upon the bearing pressures.

The four above-mentioned forces were determined at crank intervals of fifteen degrees, for all the cylinders of the engines mentioned. A glance at the diagrams on Plate I will show how the resultant pressures upon the forward and aft bearing of each cylinder were obtained. The magnitude and direction of these resultants at different crank intervals are shown on Plates II–VI.

At the top of these plates is shown also the distribution of the resultant loads upon the bearings during a revolution. In all the diagrams a full line is used to denote results for aft bearings, and a broken line the results for the forward bearings.

Table 1.

Name.	*Boston.*	*Monterey*	*New York*	*Olympia*	*Siberia*
Cylinders, inches	24, 37, 63	27, 41, 64	32, 47, 72	42, 59, 92	35, 50, 70, 100
Stroke, inches	42	30	42	42	66
Revolutions	124.8	167.6	131	141	87
Piston speed	873	848	917	987	957
Horsepower used in calculations:					
H.P.	663	856	1394	2100	2020
1st M.P.	960	837.5	1436	3090	2525
2d M.P.	...	...	...	...	1932
L.P.	1355	1162	1622	3290	1615
Total	2978	2855.5	4152	8480	8092

Plate III is for the engines of the *Monterey*, which are three-cylinder, triple-expansion naval engines. So, also, are the engines of the *New York*, whose diagrams are shown on Plate IV. Both of these engines have their cranks so arranged that the "High leads"; i. e., the order of the cranks in rotation, is H., I., L.

The diagrams for the engines of the passenger steamer *Boston* are shown on Plate II. These engines are three-cylinder, triple-expansion, with counterweights on the crank webs, so that the centrifugal force from the unbalanced portion of the cranks acts in a direction opposite to that in which it acts in the other engines, as can be seen by referring to Figs. 5 and 6, Plate I. The *Olympia* has also three-cylinder, triple-expansion engines, and the diagrams are shown on Plate V. The engines of the *Boston* and *Olympia* have their cranks so arranged that the "Low leads"; i. e., the order of the cranks in rotation is L., I., H.

The diagrams for the engines of the *Siberia* are shown on Plate VI. The engines are four-cylinder, quadruple-expansion, the arrangement of cylinders being H.P., I.P., 2d M.P., 1st M.P. The H.P. and L.P cranks make an angle of 180 degrees, as do also the 1st M.P. and 2d M.P. cranks. The L.P. and 2d M.P. cranks make an angle of 90 degrees.

The four forces mentioned above, namely, the pressure of the steam upon the piston, the weight and inertia of the reciprocating parts, the centrifugal force from the unbalanced portions of the crank webs and crank pin, the load due to the transmission through the crank webs of power from the forward cylinders, may be treated under three heads: the load through the connecting rod, the rotative force from forward cylinders, and the centrifugal force. The load through the rod is made up of the steam load upon the piston, and the inertia forces of the piston, piston rod, crosshead and that portion of the connecting rod which acts at the crosshead. The centrifugal force includes that due to the unbalanced portion of the crank webs and crank pin, and practically that due to the weight of the connecting rod acting at the crank

pin. This centrifugal force varies from 4,000 pounds in the case of the *Boston* to 92,000 pounds in the case of the *Siberia*.

The rotative force from the forward cylinders is transmitted through the crank pins, and its direction of action for a given crank position may be found by drawing a tangent to the crank-pin path at the given position. A load acting in the direction of motion at the instant will consequently be thrown upon the aft bearing of the crank, and an equal load in the opposite direction will be thrown upon the forward bearing. These loads, then, will change their direction of action through 360 degrees in a revolution.

The direction of action of the loads through the rod will vary from 15 degrees on one side of the vertical to 15 degrees on the other side.

Considering these two sets of loads, we see that when the crank is at its 0-degree and 180-degree positions the load through the rod will make an angle of 90 degrees with the direction of the load from the rotative force, while in the vicinity of the 75-degree and 285-degree positions of the crank the two loads will act in the same direction. The effect of the inertia of the reciprocating parts is to absorb the steam pressure at the beginning and end of the strokes, and to cause the load through the rod to have its maximum value when the crank is in the vicinity of the 120-degree and 300-degree positions. It should further be noticed that in the case of the aft bearing of any cylinder the load through the rod is acting in the same general direction as the load from the rotative forces, while in the case of the forward bearing, the two loads are in general opposed to one another. A glance at the figures on Plate I will make this clear.

The mean rotative force in pounds necessary for the development of one horsepower will be $p = \frac{33{,}000}{2\pi rn}$; since piston speed $= P.S. = 4rn$; $p = \frac{2 \times 33{,}000}{\pi \times P.S.} = \frac{21{,}000}{P.S.}$ (nearly). The mean force acting at the piston for the development of one

horsepower $= p_1 = \frac{33{,}000}{P.S.}$. Hence, the value of p_1 in terms of p will be, $p_1 = \frac{33}{21} p = 1.6\, p$ (nearly).

For every horsepower transmitted through a crank we should expect that a mean load of $\frac{21{,}000}{P.S.}$ pounds would come upon the adjacent bearings; while for every horsepower developed in a cylinder whose bearings we are considering, a load of $.8\frac{21{,}000}{P.S.}$ pounds would come on each bearing, as there are two bearings to take this load. It has been pointed out that these two loads make varying angles with one another, so that the resultant load would not be equal to the sum of the two loads, but would be found from formulae such as

$$L = p(HP_T - fHP_C), \quad . \quad . \quad . \quad . \quad . \quad (1)$$

for the forward bearing, and

$$L = p(HP_T + f_a HP_C), \quad . \quad . \quad . \quad . \quad . \quad (2)$$

for the aft bearing.

L = total mean load upon bearing;

$p = \frac{21{,}000}{P.S.}$;

$H.P._T$ = net horsepower transmitted from forward cylinders, or I.H.P. minus the horsepower absorbed by internal and journal friction;

$H.P._C$ = I.H.P. developed in the cylinder over the bearing;

f and f_a = coefficients whose values would be less than .8; say .55.

So far we have neglected one of the forces acting, the centrifugal force. The effect of this force is to increase the load upon the bearings, so that equations (1) and (2) might be expected to take the form,

$$L = p(HP_T - .55\, HP_C) + KC, \quad . \quad . \quad . \quad (3)$$

for forward bearings; and for aft bearings,

$$L = p\,(HP_r + .55\,HP_c) + KC, \quad . \quad . \quad . \quad (4)$$

where C = the centrifugal force, and K = coefficient less than 1. It was found, however, that better results could be obtained by increasing the coefficient of HP_c to allow for the increased load due to the centrifugal force. This increase in most cases will cause the coefficient in (1) to be positive rather than negative, so that we can write one expression for the load on all bearings,

$$L = p\,(HP_r + fHP_c), \quad . \quad . \quad . \quad . \quad . \quad (5)$$

where f will have a value greater than $-.55$ for forward bearings, and greater than .55 for aft bearings.

As we go aft along the engine the load upon the bearings due to the transmitted horsepower increases, while the load due to the centrifugal force is constant in most cases. We should, therefore, expect that as the centrifugal force becomes relatively of less importance, that the values of f would approach the values, .55 and $-.55$.

The values of f for the bearings of the engines investigated are shown in Table 2, and are also plotted upon Plate VII.

It will be seen from the curves on Plate VII that the values of the coefficients for the aft bearings seem to be approaching a value less than .6 as a limit, and the curves are gradually flattening out as we go aft along the engine. The values of the coefficients for the forward bearings are also becoming rapidly negative; the curve for the fourth cylinder of the Quadruple promising to be almost entirely negative.

The directions of the curves passing through the points for the quadruple engines of the *Siberia* were assumed, as it was impossible to get from engine builders the small amount of data necessary for a more complete determination of the curves for this class of engine. However, I believe that the coefficient for any bearing of a quadruple-expansion engine will be greater than the coefficient for the same bearing of a triple-expansion engine, both bearings, of course, being subjected to the same centrifugal force. As the reciprocating parts

Table No. 2.

Name.	*Boston*	*Monterey*	*New York*	*Olympia*	*Siberia*
Centrifugal force (pounds), 1st cylinder . . .	500	...	...	...	93,275
2d cylinder . . .	—1700	22,000	39,000	79,000	93,275
3d cylinder . . .	—4100	...	...	...	103,500
4th cylinder . . .	...	...	...	...	103,500
Value of p	24.05	25.08	22.9	21.28	21.95
1st cylinder, $L = p\,HP_c f_1$ . . . . . $f_1 =$	(.865) .836	.741	.833	1.182	1.385
2d cylinder, forward, $L = p(HP_T + f_2 HP_c)$. $f_2 =$	(.138) .309	.10	.119	.369	.662
aft, $L = p(HP_T + f_3 HP_c)$. $f_3 =$	(.63) .638	.583	.616	.81	1.096
3d cylinder, forward, $L = p(HP_T + f_4 HP_c)$. $f_4 =$	(—.095) —.075	—.138	—.112	.149	.433
aft, $L = p(HP_T + f_5 HP_c)$. $f_5 =$	(.67) .615	.63	.66	.772	.91
4th cylinder, forward, $L = p(HP_T + f_6 HP_c)$. $f_6 =$	...	...	...	...	.113
aft, $L = p(HP_T + f_7 HP_c)$. $f_7 =$	...	...	...	...	.84
Mean load on bearings of 1st cylinder, . . .	13,340	15,900	25,550	52,850	57,000
" " " forward bearing of 2d cylinder .	23,100	23,600	35,800	68,900	81,000
" " " aft bearing of 2d cylinder . .	30,600	33,750	52,100	97,750	105,200
" " " forward bearing of 3d cylinder .	36,600	38,400	60,750	120,700	118,000
" " " aft bearing of 3d cylinder . .	59,000	60,800	89,250	164,400	138,000
" " " forward bearing of 4th cylinder .	...	...	...	...	146,000
" " " aft bearing of 4th cylinder . .	...	...	...	...	172,000

of a quadruple-expansion engine are generally lighter than those of a triple-expansion engine of the same power, the quadruple-expansion engine whose bearings were subjected to the same centrifugal force as those of a Triple, would probably be of a slightly greater power than the Triple. At a given bearing in the Quadruple, however, only 75 per cent. as much power has been developed in the forward cylinders, as has been developed forward of the same bearing in the Triple; therefore the effect of the same centrifugal force in the Quadruple would be greater than in the Triple. Again, the effect of the centrifugal force upon a bearing is expressed as a certain percentage of the load coming through the rod. In a Quadruple the power developed in a cylinder is 75 per cent. of the power developed in a cylinder of a Triple of equal total power; therefore the percentage in a Quadruple would have to be larger than in a Triple in order to allow for the effect of equal centrifugal forces.

If the crank webs of the *Boston* had no counterweights attached to them the value of the centrifugal force would be about 43,600 pounds. The values given in the parentheses of Table I under the heading *Boston* are taken from the curves for that value of centrifugal force, and show what the coefficients would be if the engine were without counterweights. It will be seen that the effect of the counterweights is to decrease the load on the H.P. and aft L.P. bearings and to increase it slightly on the others.

The value of the centrifugal force acting upon the bearings of an engine may be readily calculated if the dimensions of the crank web and pin, and the weight of the connecting rod are known. It may be obtained approximately by the following formulae:

Centrifugal force of crank webs and pin =

$$C_1 = fW_1n^2S, \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

where

$f = .55$ for solid-forged cranks, and .61 for built-up cranks;

W_1 = weight of unbalanced portion of crank webs and pin;

n = revolutions per second;

S = stroke of engine in feet.

The centrifugal force of the crank end of the connecting rod will be given very closely by the formula,

$$C_2 = .375\ W_2 n^2 S, \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

where W_2 = total weight of connecting rod, and n and S are as above.

$C_1 + C_2$ = total centrifugal force, and will locate the part of the curves from which the coefficients for the various bearings may be taken.

In cases where it is not convenient to calculate the centrifugal force, or where rough approximations to the mean loads upon bearings are desired, the values of the coefficients given on Plate VIII may be used. The curves on this plate give the correction that should be applied to the mean load obtained by the formula if the centrifugal force were known. It will be seen that in the case of all bearings except those of the 1st Cylinder, the correction is a negative quantity varying from 2,000 to 4,000 pounds for centrifugal forces of 60,000 pounds and less. A centrifugal force of 60,000 pounds would probably be found in an engine of about 5,000 I.H.P., so that these values of coefficients could be used for the majority of engines.

The diagrams at the top of Plates II–VI show the distribution of loads upon the bearings throughout the stroke. It will be noticed in the case of the first cylinders that as the centrifugal force increases the points of application for the loads are more uniformly distributed around the bearing. The *Boston*, with a centrifugal force of 500 pounds on the H.P. bearings, has the loads acting at practically two points;

while the *Siberia*, with a centrifugal force of 92,000, has the loads well distributed around the first cylinder bearings.

The diagrams for the other bearings show that the loads are quite uniformly distributed around the surface of the aft bearings, with a slight tendency to group at the top and bottom of these bearings. The maximum loads are directed towards the top and bottom of the bearings in the case of the aft bearing of the 2d Cylinder, while they are directed towards the sides in the case of the aft bearing of the 3d Cylinder. The loads on the forward bearings show a tendency to group themselves at the sides of the bearings. Although the load upon a forward bearing is less than that upon an aft bearing, this tendency of the load to act upon the sides, in the case of those bearings which are largely cut away, may cause the pressure per square inch upon the forward bearing to equal that upon the aft bearing.

In obtaining the above results we have neglected the friction of pistons, piston valves, stuffing boxes, and the weight and inertia of the valve gear.

Figs. 9 to 12 on Plate VI show the value of these neglected items as worked out for the *Siberia*, upon the following assumptions :

1. The horsepower absorbed by friction of piston rings, piston valves, and stuffing boxes equals 8 per cent. of total I.H.P.
2. The coefficient of friction will vary as the square root of the speed of rubbing.
3. The steam pressure has access to the back of the rings so that the load upon the rings decreases as the steam pressure decreases.
4. The coefficent of friction varies inversely as the temperature and consequently increases as the steam pressure decreases.
5. Assumptions 3 and 4 neutralize one another.

We have, then, the frictional load in any piston rod or valve stem varying as the square root of the speed of rubbing, and directly as the area of rubbing surface.

The pistons have a speed of $2 \times 87 \times 5.5 = 957$ feet per minute, and the valves have a speed of $2 \times 87 \times .833 = 145$ feet per minute. The H.P. and 1st M.P. have each three piston rings with $\frac{5}{8}$-inch rubbing faces, and the 2d M.P. and L.P. have two rings with $\frac{3}{4}$-inch rubbing faces.

The piston valves have a single solid ring about 6 inches deep, with provision made for splitting and expanding after two or three years' service. It was assumed, however, that the rings would be pressing against the valve liner with a pressure per unit area equal to the steam pressure in the receiver. The width of the rubbing surface was assumed to be equal to the width of the solid ring, or 6 inches at each end.

The cylinder diameters are 35 inches, 50 inches, 70 inches, 100 inches. Taking as a unit the horsepower absorbed by the H.P. piston with three $\frac{5}{8}$-inch rings, moving at a piston speed of 957 feet per minute, we have—

Horsepower absorbed by

H.P. piston rings, etc.	$= 1.00x$
1st M.P. piston rings, etc. $= \frac{50}{35}x$ . . .	$= 1.43x$
2d M.P. piston rings, etc. $= \frac{70}{35} \cdot \frac{1.5}{1.875}x$.	$= 1.60x$
L.P. piston rings, etc. $= \frac{100}{35} \cdot \frac{1.5}{1.875}x$.	$= 2.29x$
Total horsepower absorbed in cylinders .	$= 6.32x$

The diameters of the piston valves are as follows:

H.P. piston valve, ...	(1) 20 inches.
1st M.P. piston valves, ...	(2) 17 inches.
2d M.P. piston valves, ...	(2) 28 inches.
L.P. piston valves, ...	(4) 25 inches.

These valves have a stroke of 10 inches while the pistons have a stroke of 66 inches, so that the valves have a speed of 145 feet per minute, and the pistons a speed of 957 feet per minute. The coefficent of friction for the valves will be less than that for the piston rings in the ratio of $\sqrt{957/145} = 2.57$.

Frictional horsepower absorbed by

H.P. piston valve $= \frac{20}{35} \cdot \frac{8.5}{1.875} \cdot \frac{10}{66} \cdot \frac{1}{2.57} x \cdot = .267x$

1st M.P. piston valve $= \frac{34}{20} \cdot 2.67x \ldots = .45x$

2d M.P. piston valve $= \frac{56}{20} \cdot 2.67x \ldots = .748x$

L.P. piston valve $= \frac{100}{20} \cdot 2.67x \ldots = 1.335x$

Total horsepower absorbed by piston-valve friction, $\ldots = 2.800x$

Total frictional horsepower $= 9.12x = .08 \times 8{,}090 = 648$

Hence, $x = 71$.

Frictional horsepower absorbed by

H.P. piston = 71, load in piston rod = 2,450 pounds.
1st M.P. piston = 101.5, load in piston rod = 3,500 pounds.
2d M.P. piston = 113.5, load in piston rod = 3,915 pounds.
L.P. piston = 163, load in piston rod = 5,625 pounds.
H.P. piston valve rings = 19, load in valve stem = 4,320 lbs.
1st M.P. do. = 32, load in valve stems = 7,280 lbs.
2d M.P. do. = 53.2, load in valve stems = 12,100 lbs.
L.P. do. = 94.8, load in valve stems = 21,550 lbs.

In some offices it is customary to assume the frictional load in valve stems as equal to four times the weight of valve, valve stem, crosshead and block. The ratio of the frictional loads given above to the weight of the items mentioned, is for the H.P., 2.81; for the 1st M.P., 2.67; for the 2d M.P., 2.57; or the L.P., 2.58. It will be seen from this that the frictional loads used in this computation are less than those obtained in the usual way.

The frictional loads in valve stems and piston rods thus obtained were combined with the forces due to weight, inertia and pressure upon balance pistons. Figs. 9–12, Plate VI, show the resultant of these forces, which for all practical purposes act vertically upward or downward. Fig. 9 is typi-

cal of valves taking steam in the middle; the eccentric being then about 50 degrees behind the crank. Fig. 12 is typical of valves taking steam on the ends; the eccentric being then about 130 degrees ahead of the crank.

The general effect of the friction and of the valve gear can be predicted from a consideration of the conditions. The friction of the piston rings will neutralize a certain portion of the total load from the steam pressure, so that the load through the connecting rod will be less than we have taken it. If the valve takes steam at the middle, the eccentric will be about 50 degrees behind the crank, and the friction of the valve gear will, in general, introduce a load opposed to the load through the rod. If the valve takes steam at the ends, the eccentric will be about 130 degrees ahead of the crank, and the friction of the valve will, in general, introduce a load acting in the same direction as the load through the rod. So we should expect that a valve taking steam at the middle would cause the load upon the adjacent bearing to be less than we have figured, while a valve taking steam at the ends would cause an increase in the load.

It can be seen that when the forces shown in Fig. 9 are combined with the proper forces in Fig. 1, the mean resultant load upon the bearing will be decreased. On the contrary, when the forces shown in Fig. 12 are combined with the proper forces in Fig. 4, the mean resultant load on the bearing will be increased.

In the engine of the *Siberia* the H.P. valve is forward of the cylinder and affects the forward bearing of that cylinder. The ordinates to the dotted curve in Fig. 9 give the values of the additional loads due to the valve gear and friction, and the effect of these forces when combined with those of Fig. 1 is to reduce the mean load upon the bearing by 5 per cent.

The L.P. cylinder comes next to the H.P. cylinder, so that the aft bearing of the H.P. cylinder carries seven-eighths of the load due to the weight and friction of two 25-inch valves. The L.P. valves take steam on the outside, but as the crank is 180 degrees away from the H.P. crank the curve of addi-

tional loads for the aft H.P. bearing is the same as that for a gear taking steam in the middle. The effect of this gear is to decrease by 12 per cent. the mean resultant load previously obtained. This latter is, of course, an extreme case, as it is seldom that one of the bearings of a 35-inch cylinder has to carry the load from two 25-inch valves.

Fig. 10 shows that very little of the effect of the L.P. gear is carried by the L.P. bearings; it is thrown largely upon the bearings of the H.P. and 2d M.P. The effect of the valve gear is to increase the load on the forward and aft bearings by about 1 per cent.

The 2d M.P. valve takes steam on the inside, and one-half the load from this gear is carried by the aft bearing, with the result that the mean load previously obtained is decreased by 4 per cent. The effect of the L.P. gear upon the forward bearing is to decrease the resultant load by about 2 per cent.

The 1st M.P. valve takes steam on the outside, and the load from the gear is carried by the aft bearing, with the result that the mean load previously obtained is increased 4 per cent. The forward bearing, carrying half the load of the 2d M.P. valves, has its mean load decreased by 2 per cent.

It can be seen from this that when the mean load upon a bearing is calculated neglecting the effect of friction and of the valve gear, a reduction should be made if the bearing is subjected to a curve of additional loads similar to Fig. 9, Plate VI, and an addition should be made if the curve is similar to Fig. 12. The amount of reduction or addition in the case of aft bearings will vary from 4 to 6 per cent. As the additional loads in the case of forward bearings are acting in a direction perpendicular to the general direction of loads upon the bearing the resultant will be but slightly affected in amount, the principal effect being upon the direction, causing the loads to act more, or less, upon the sides of the bearing.

The assumptions upon which these results are based are, of course, arbitrary. It was thought best, however, to make a set of assumptions and find the resulting effects; then the

effect of any different assumptions could be seen more readily. The frictional load in the piston rod of the forward cylinders is probably too small, a sufficient allowance not having been made for the friction of the stuffing boxes. The frictional loads in the valve stems are probably too large for a set of gears having restrained piston-valve rings. However, if some of the load in the valve stems is transferred to the piston rods, the curve of additional loads for a valve taking steam in the middle will remain practically the same, while the value of the ordinates of the curve for a valve taking steam at the ends will be decreased. The effect of increasing or decreasing the horsepower assumed to be used up in friction can be readily seen.

It would appear, then, that no very great error would be introduced by neglecting the effect of the valve gear in calculating the loads upon the bearings, and dealing only with the horsepower transmitted from the forward cylinders and developed in the cylinder over the bearing in question. If, however, the effect of the valve gear is to be taken into account, the approximate mean load upon a bearing can be obtained by use of a formula such as,

$$L = p\,(HP_T + fHP_C)f_V, \quad . \quad . \quad . \quad . \quad (8)$$

Where

$$p = \frac{21{,}000}{P.S.},$$

HP_T = net horsepower transmitted from forward cylinders;
f = coefficient obtained from curves on Plate VII;
HP_C = horsepower developed in cylinder over the bearing;
f_V = coefficient greater or less than 1, dependent upon the character of the curve of additional loads.

The question of the probable amount and distribution of pressure upon a common bearing between two cylinders was also investigated. The loads acting upon the two bearings to be united were compounded at the proper angles and the resultants found. The factor by which the sum of the loads upon the two bearings united is to be multiplied to get the

load on the common bearing, is given in each case in the tables below.

Table 3.

Name.	Type of engine.	Angle between cranks.	Factor.	Load.
		degrees.		*pounds.*
	Common Bearing between 1st and 2d Cylinders.			
Monterey ...	Triple—High leading.....	120	.623	24,600
New York..	Triple—High leading.....	120	.547	33,550
Siberia	Quadruple.....	180	.435	60,000
Boston.....	Triple—Low leading.....	240	.86	31,350
Olympia.....	Triple—Low leading.....	240	.81	98,250
	Common Bearing between 2d and 3d Cylinders.			
Siberia	Quadruple.....	90	.47	100,000
Monterey ...	Triple—High leading.....	120	.64	46,200
New York..	Triple—High leading.....	120	.595	67,100
Boston.....	Triple—Low leading.....	240	.877	58,900
Olympia.....	Triple—Low leading.....	240	.918	200,800
	Common Bearing between 3d and 4th Cylinders.			
Siberia.....	Quadruple	180	.873	248,500

It will be noticed in the case of the quadruple engine that the factor for the common bearing between the 1st and 2d cylinders, where the cranks make an angle of 180 degrees, is .435, while in the common bearing between the 3d and 4th cylinders, where the cranks also make an angle of 180 degrees, the factor is .873. The difference is due to the greater relative influence of the centrifugal force in the first case, causing the load upon the aft bearing of the 1st cylinder and the load upon the forward bearing of the 2d cylinder to unite at a mean angle greater than 90 degrees. In the case of the bearing between the 3d and 4th cylinders the two loads unite at a mean angle of about 30 degrees.

When two cranks making an angle of 180 degrees have a common bearing between them, the centrifugal force will, of course, be eliminated if the parts are of equal weight; and the loads through the rod will balance one another if equal powers are developed in the two cylinders. We ought, therefore, to be able to get the load upon the common bearing from the

horsepower being transmitted through the cranks. In this instance the horsepowers being developed in the cylinders are, 2,020 in the 1st, 2,525 in the 2d, 1,932 in the 3d and 1,615 in the 4th. If we consider a common bearing between the 3d and 4th cylinders, and take the cranks when the 3d is at its 90-degree position, since the common bearing is acting as the aft bearing of the 3d cylinder it will be subjected to a downward load due to the horsepower (2,020+2,525 = 4,545) being transmitted from the 1st and 2d cylinders. Since the bearing is also acting as the forward bearing of the 4th cylinder, there will be in addition a downward load due to the transmission through the 4th crank pin of the power (2,020+2,525+1,932 = 6,477 H.P.) from the 1st, 2d and 3d cylinders. The horsepower developed in the 3d cylinder is greater by 315 than that developed in the 4th, so there would be a further load upon the bearing. The total load upon the common bearing will then be

$$21{,}000/PS\,(4{,}545+6{,}477+.8\times 315)=21.95\times 11{,}274$$
$$=247{,}000 \text{ pounds.}$$

The mean load upon the common bearing obtained by combining the loads upon the two bearings at the proper angle is 248,500.

On Plates IX and X will be found diagrams showing the distribution of the points of application of the loads upon the common bearings during a revolution. It will be noticed from Table 3 that the High Leading engines have smaller mean loads upon the common bearings, but from these diagrams it will appear that these loads act upon the sides of the bearings; while the Low Leading engines with the much greater mean loads have their common bearings acted upon more at the top and bottom. It will also be evident from the diagrams that in the case of the common bearing between the 1st and 2d cylinders of the Quadruple, and of the High Leading engines, that the load is acting mainly upon the bottom of the bearing, and I may add that during the short time in which it is acting upon the top the load is very small.

Where there is a common bearing between the 2d and 3d cylinders of the High Leading engines the sides of the bearings receive most of the loads, while in the Low Leading engines and Quadruple the loads are acting at an angle of about 45 degrees with the vertical.

The forces producing loads upon the crank pin are, the pressure of the steam modified by the inertia of the reciprocating parts, and the centrifugal force of the crank end of connecting rod. The mean load upon the crank pin may be obtained approximately from the horsepower developed in the cylinder. It has been shown that the mean force acting throughout the stroke of the piston to develope one horsepower is $1.6p$, where $p = 21{,}000/PS$. We should, therefore, expect that the mean crank-pin load would be given by a formula

$$L_1 = 1.6 \times I.H.P. \times p$$

where L_1 = mean load on crank pin;

$I.H.P.$ = indicated horsepower developed in cylinder;

$p = 21{,}000/P.S.$

It was found, however, that the coefficient had a value of 1.41 when the effective pressure upon the piston at the beginning of the stroke was just about sufficient to accelerate the parts, and the value increased as the effective pressure was greater or less than this. The coefficient was also affected by the distribution of pressure throughout the stroke, the centrifugal force of the crank end of the connecting rod, and the pounds of reciprocating parts per horsepower developed.

Table 4 shows the variations of the factor with some of these causes.

It will be seen that by making $f = 1.6$ we shall be using about a mean of the coefficients there given.

In Table 5 are given the pressures upon the crank pin and main bearings of the engines investigated.

It will be noticed that, except in the cases of the bearings of 1st cylinder and the forward bearings of the 2d cylinder, the mean pressure upon the bearing exceeds the mean pressure upon the crank pin. The mean crank-pin pressure

Table 4.

		Cent. throw of crank end of connecting rod.	Pounds of reciprocating parts per I.H.P. in cyl.	Coefficient	Mean load on crank pin.
Monterey.....	H. P.	12,000	3.5	1.43	30,650
	M. P.	12,000	4.83	1.41	29,550
	L. P.	12,000	4.58	1.52	44,300
New York....	H. P.	20,500	4.25	1.53	49,100
	M. P.	20,500	4.56	1.47	48,300
	L. P.	20,500	5.02	1.59	59,000
Boston.....	H. P.	22,000	10.2	2.24	35,700
	M. P.	22,000	7.5	1.67	38,550
	L. P.	22,000	6.4	1.63	53,000
Siberia.....	H. P.	32,375	6.86	1.47	65,000
	1st M. P.	32,375	9.68	1.75	62,100
	2d M. P.	32,375	9.14	1.41	59,800
	L. P.	32,375	8.83	1.82	101,000
Olympia.....	H. P.	43,400	5.3	1.69	75,500
	M. P.	43,400	3.97	1.48	99,700
	L. P.	43,400	4.57	1.86	130,000

Table 5.

		Crank-pin bearing surface.	Mean pressure on crank pin.	Maximum pressure on crank pin.	Ratio of maximum to mean.	Area of main bearing surface.	Mean pressure on main bearing.		Maximum pressure on main bearing.		Ratio of maximum to mean.	
							Fwd.	Aft.	Fwd.	Aft.	Fwd	Aft.
		Sq. ins.	*Lbs.*	*Lbs.*		*Sq. ins.*	*Lbs*	*Lbs.*	*Lbs.*	*Lbs.*		
Monterey.....	H.P.	165	186	333	1.8	132	121	121	242	242	2	2.
	M.P.	165	179	340	1.9	132	179	256	349	446	1.95	1 69
	L.P.	165	269	510	1.9	132	291	461	485	648	1.67	1.41
New York.....	H.P.	229	213	365	1.71	189	138	138	270	270	2	2.
	M.P.	229	210	387	1.82	189	189	276	360	468	1 9	1.7
	L.P.	229	256	488	1.9	189	321	472	567	649	1.76	1.35
Boston.....	H.P.	195	184	343	1.88	169	79	79	187	187	2.38	2.38
	M.P.	195	198	400	2.02	169	137	181	177	361	1.3	1.99
	L.P.	195	272	524	1.93	169	216	349	303	452	1.39	1.3
Olympia.....	H.P.	365	206	448	2.18	316	168	168	324	324	1.04	1.94
	M.P.	365	272	541	1.99	316	218	310	374	528	1.72	1.71
	L.P.	365	355	650	1.84	316	381	521	675	695	1.77	1.34
Siberia.....	H.P.	389	167	318	1.91	390	142	142	246	246	1.70	1 70
	L.P.	389	259	498	1.92	390	208	270	316	487	1.53	1.80
	2d M.P.	389	153	333	2.17	F. 390 A. 449	303	308	534	530	1.76	1 72
	1st M.P.	389	159	318	2.0	F. 449 A. 508	326	339	578	485	1.78	1 43
Mean.....		...	221	...	1.93	...	213	275	...	...	1.66	1.72
Mean of 1st cylinders.....		...	...	...	...	...	130	130	...	...	...	...
2d cylinders.....		...	...	...	...	...	186	258	...	...	...	...
3d cylinders.....		...	...	...	...	...	302	467	...	...	...	...

ranges from 150 pounds to 355 pounds per square inch, while the mean pressure upon the forward bearings ranges from 79 pounds to 381 pounds per square inch, and on the aft bearing from 79 pounds to 521 pounds per square inch.

Assuming that the present sizes of crank pins and main bearings have been dictated by experience, it would appear that the crank pin cannot carry as great a mean pressure as some of the aft bearings.

It is probable that the aft main bearing of the aft cylinder can carry a greater mean pressure per square inch than any other, due to the fact that the load is well distributed over the whole bearing during a revolution, so that it is more easily kept cool. Table 6 shows the mean pressure per square inch upon the aft bearing of the aft cylinder during the trial ships of certain boats.

Table 6.

Name.	Rev.	Piston speed.	$2\frac{1,000}{P.S.}$	Total	Load on aft bearing of aft cylinder	Pressure per square inch.	Remarks.
			pounds.	*I.H.P.*	*pounds.*	*pounds.*	
DeLong	357	1,071	19.6	1,512	26,650	438	
Whipple	317	1,162	18.1	3,875	63,000	456	
Truxtun	311	1,140	18.4	3,821	63,100	457	
Worden	324	1,185	17.8	4,148	65,800	477	
Perry	373	1,368	15.3	4,577	61,600	518	Bearing heated.
Hull	412	1,236	17.	4,635	69,700	690	
Hopkins	373	1,119	18.8	4,463	74,300	737	
Cleveland	186	932	22.5	2,293	48,000	472	
Tacoma	196	977	21.5	2,605	50,100	493	
Helena	280	840	25.	987	21,150	495	
Annapolis	147	685	30.6	1,208	31,500	540	
Massachusetts.	133	931	22.5	5,078	101,000	415	
Iowa	109	870	24.2	5,949	124,000	430	
Maine	127	890	23.6	7,734	155,500	470	
Kentucky	114	912	23.0	6,069	120,000	495	
Ohio	124	991	21.2	6,060	152,700	580	4 cylinder.
Colorado	126	1,008	20.9	13,350	249,000	610	4 cylinder.
New York	135	944	22.2	8,636	168,500	690	

In only one instance is any heating reported, so that it would seem as if the pressure could be carried up to 700 pounds per square inch with the lubrication usual on trial trips.

In certain experiments reported in the "Proc. Inst. M. E.," May, 1888, it was found that with a cylindrical journal heating would occur with a bearing pressure of 100 pounds per square inch when an inefficient method of lubrication was used, but that with good lubrication pressures of 600 pounds per square inch could be safely carried without heating.

It would therefore appear to be safe, with the lubrication ordinarily employed, to allow a mean pressure of from 300 pounds to 400 pounds per square inch upon the aft bearings of all cylinders except the 1st. In the case of the bearings of the 1st cylinder and the forward bearings of the others, where the load is often acting upon a limited portion of the bearing, and where there is a sudden reversal of the direction of the load, it would probably be better to allow from 200 pounds to 300 pounds per square inch pressure. The loads upon the crank pins are about the same in character as those upon the bearings of the 1st cylinder, and the same pressure per square inch could be used.

I wish to acknowledge my indebtedness to the Newport News Shipbuilding Company for the data of the engines of the Pacific Mail Steamship Company's steamer *Siberia*, and to the Fore River Shipbuilding Company for the data of the engines of the Fall River Line steamer *Boston*.

AO - CRANK.
AB - LOAD IN ROD.
BC - CROSS THROW OF ROD.
CD - INERTIA + WEIGHT
DE = CENTRIFUGAL THROW OF CRANK
EF - LOAD ON AFT BEARING FROM ROTATIVE FORCE OF FORWARD CYLINDERS.
EG - REACTION ON FORWARD BEARING
AE - RESULTANT ON HP BEARINGS
AF = " " AFT " OF MP+LP
AG = " " FOR'D " " " "

FIG. 1
HP - 45°
NEW YORK

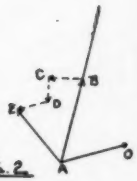


FIG. 2
HP - 255°
NEW YORK

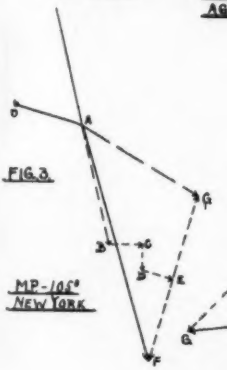


FIG. 3
MP - 105°
NEW YORK

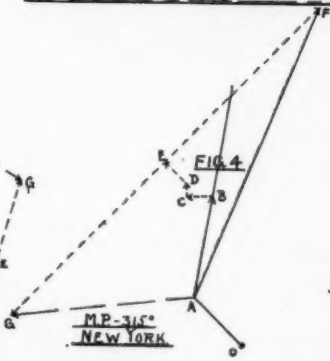


FIG. 4
MP - 315°
NEW YORK

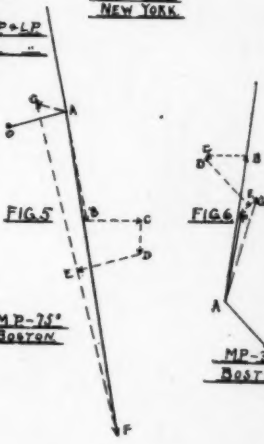


FIG. 5
MP - 75°
BOSTON

FIG. 6
MP - 315°
BOSTON

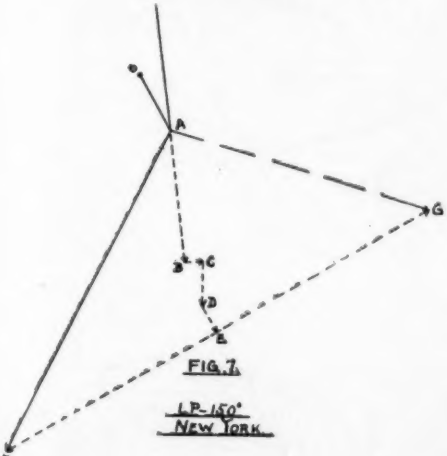


FIG. 7.
LP - 150°
NEW YORK

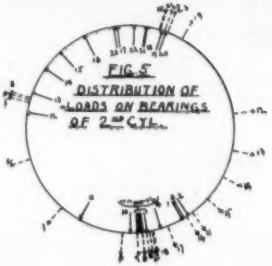
FIG. 8.
LP - 285°
NEW YORK

Plate I.

Diagrams showing Intensity, Direction and Distribution of Loads on the main Bearings of the Engines of the S. S. *BOSTON*.

Full lines are for After Bearings of Cylinders. Broken lines are for Forward Bearings of Cylinders.

0 ——— 20,000 lbs.



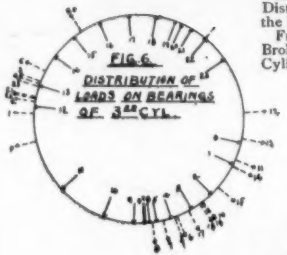
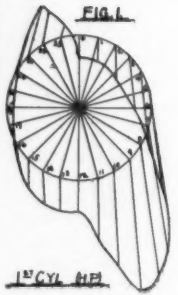
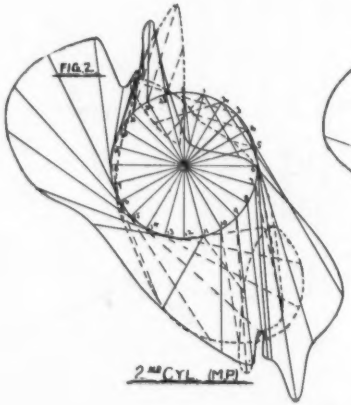
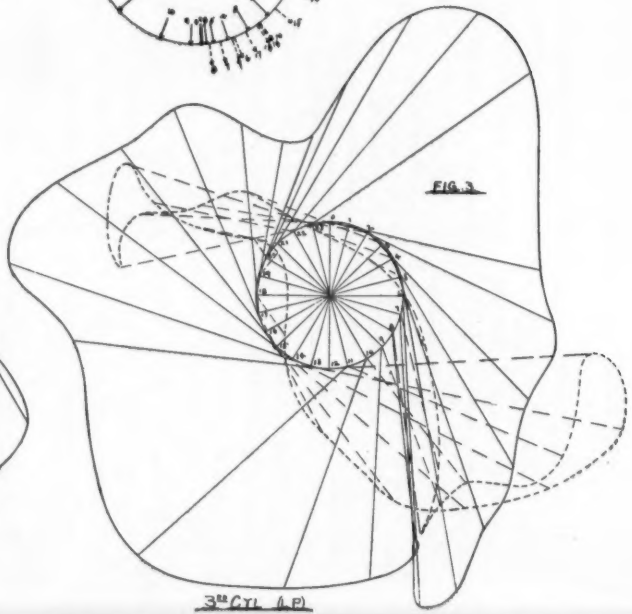


Plate II.







Diagrams showing Intensity, Direction and Distribution of

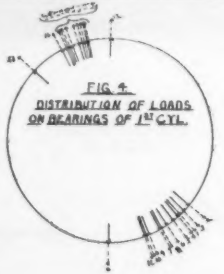
Plate III.

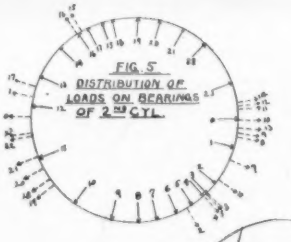
Diagrams showing Intensity, Direction and Distribution of Loads on the main Bearings of the engines of U. S. S. *MONTEREY*.

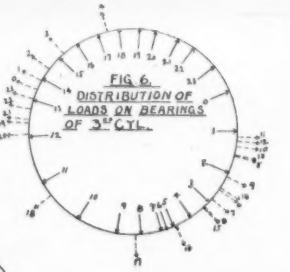
Full lines are for After Bearings of Cylinders.

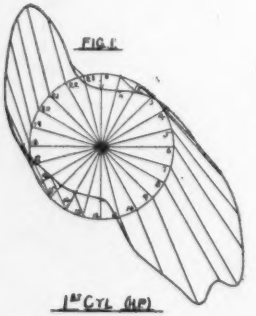
Broken lines are for Forward Bearings of Cylinders.

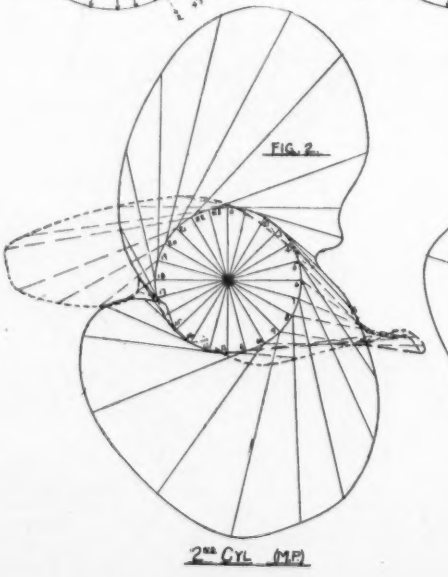
0 ——— 20,000 lbs.











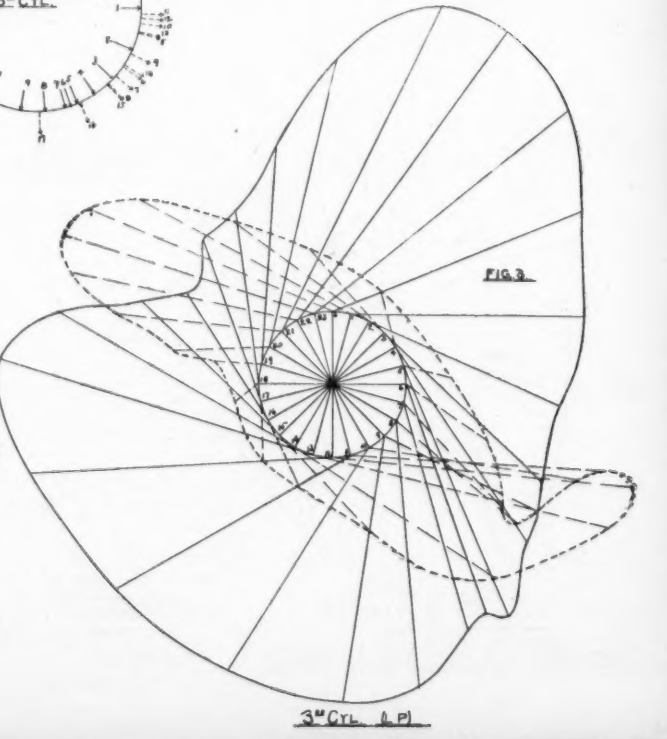


Plate IV.

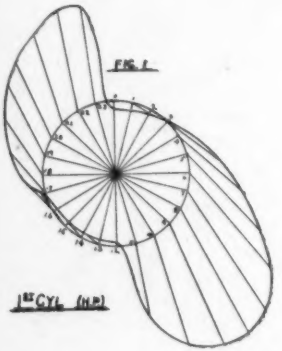
FIG 4
DISTRIBUTION OF LOADS ON BEARINGS OF 1ST CYL.

FIG 5
DISTRIBUTION OF LOADS ON BEARINGS OF 2ND CYL.

FIG 6.
DISTRIBUTION OF LOADS ON BEARINGS OF 3RD CYL.

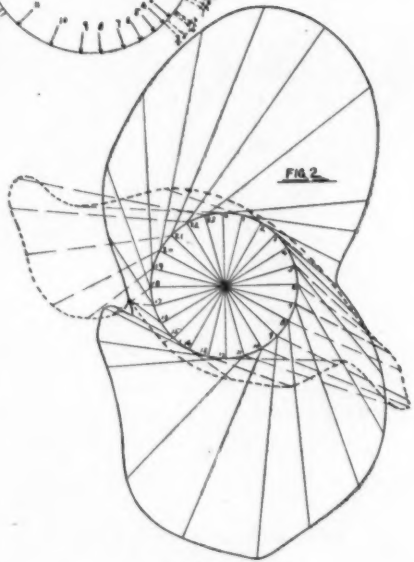
DIAGRAMS SHOWING INTENSITY, DIRECTION AND DISTRIBUTION OF LOADS ON THE BEARINGS OF THE ENGINES OF
USS NEW YORK.
FULL LINES ARE FOR AFT BEARINGS OF CYLINDERS
BROKEN - - - - FORE " " " "

FIG 1



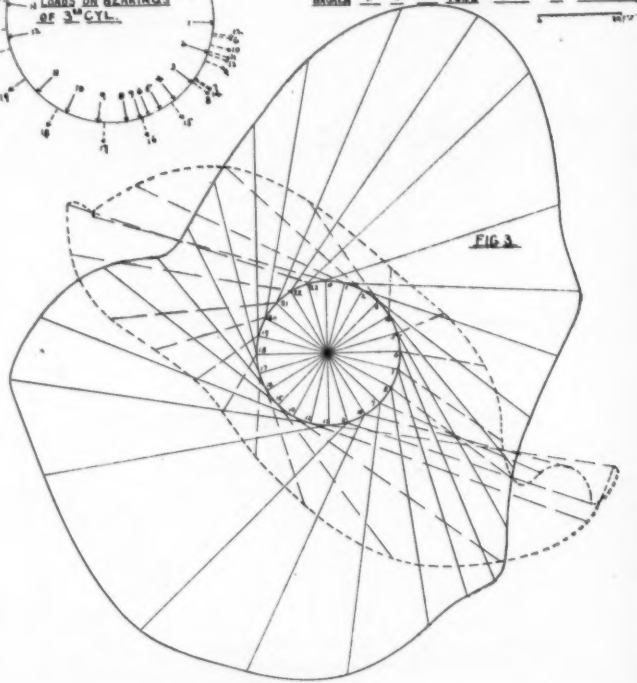
1ST CYL (H.P.)

FIG 2



2ND CYL (M.P.)

FIG 3



3RD CYL (L.P.)

2ND CYL (MP)

3RD CYL (LP)

FIG 4
DISTRIBUTION OF LOADS ON BEARINGS OF 1ST CYL.

FIG 5
DISTRIBUTION OF LOADS ON BEARINGS OF 2ND CYL.

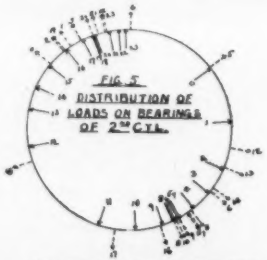


FIG 6
DISTRIBUTION OF LOADS ON BEARINGS OF 3RD CYL.

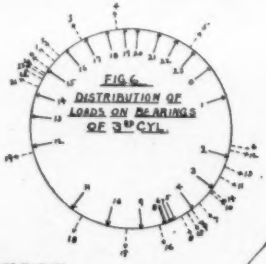


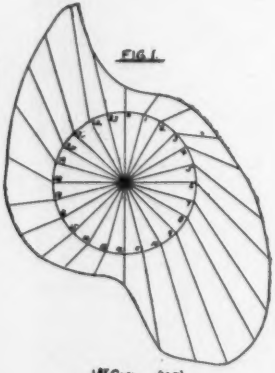
Plate V.

DIAGRAMS SHOWING INTENSITY, DIRECTION AND DISTRIBUTION OF LOADS ON THE MAIN BEARINGS OF THE ENGINES OF

USS OLYMPIA

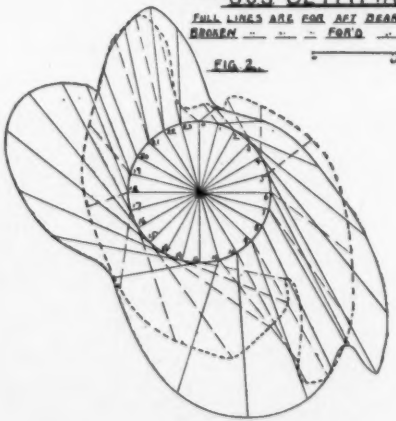
FULL LINES ARE FOR AFT BEARINGS OF CYLINDERS.
BROKEN " " " FOR'D " " " "

FIG 1.



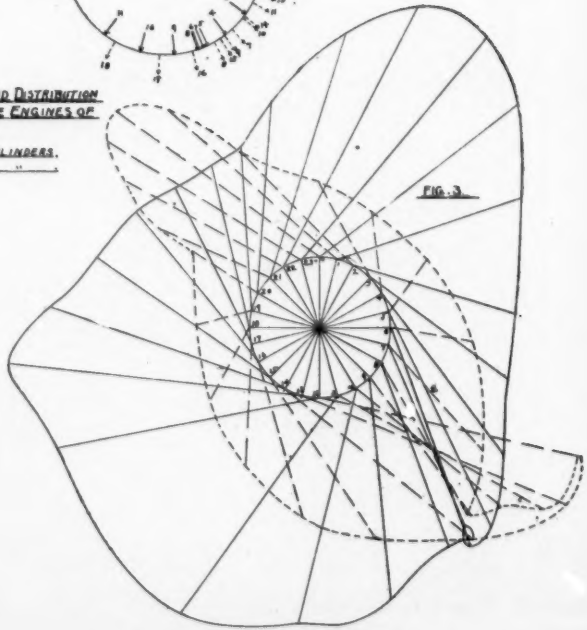
1ST CYL (HP)

FIG 2.



2ND CYL (MP)

FIG 3.



3RD CYL (LP)

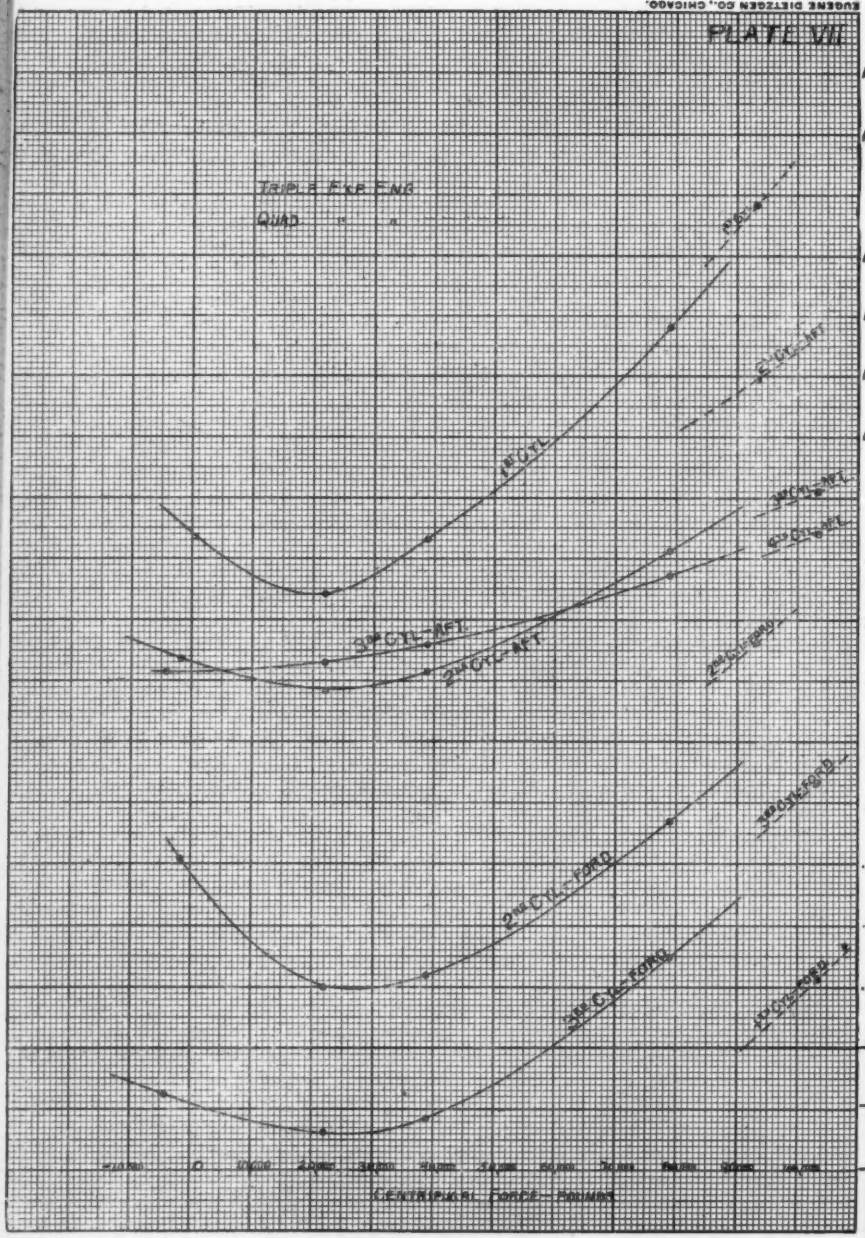


PLATE VII.
TRIPLE EXP. ENG.
QUAD. " "
1ST CYL.
2ND CYL. FORE
2ND CYL. AFT
3RD CYL. FORE
3RD CYL. AFT
1ST CYL. FORE
1ST CYL. AFT
2ND CYL. FORE
2ND CYL. AFT
3RD CYL. FORE
3RD CYL. AFT
CENTRIFUGAL FORCE - POUNDS
COEFFICIENTS FOR BEARINGS.
EUGENE DIETZGEN CO., CHICAGO.

Plate VII.

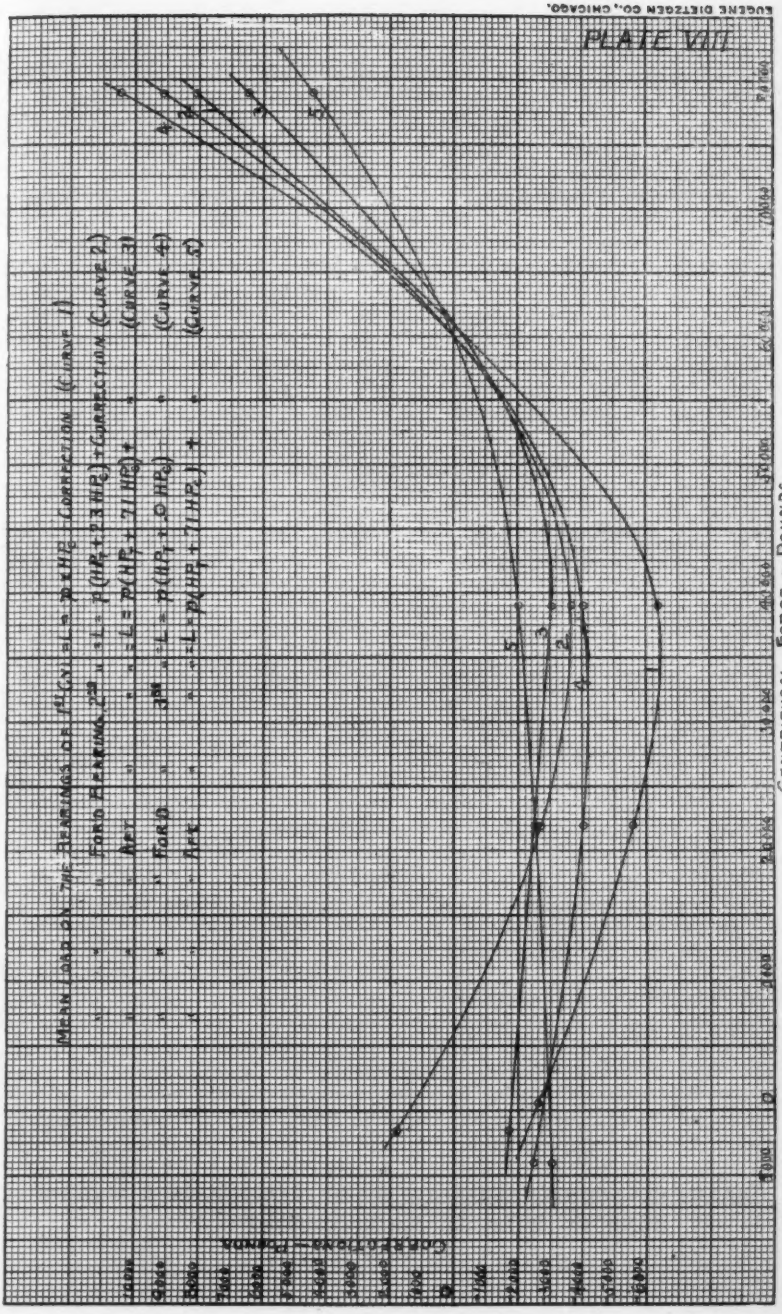


PLATE VIII
EUGENE DIETZGEN CO., CHICAGO.
MEAN LOAD ON THE BEARINGS OF F(CYL.) L = 2πKH_E CORRECTION (CURVE 1)
FORD BEARING .250 " " L = P(HP + 2.3 HP_E) + CORRECTION (CURVE 2)
" " " " L = P(HP + 7.1 HP_E) + " (CURVE 3)
ARP " " " L = P(HP + 0 HP_E) " (CURVE 4)
FORD " .312 " L = P(HP + 0 HP_E) " (CURVE 5)
ARP " " " L = P(HP + 7.1 HP_E) + "
CORRECTIONS - POUNDS
CENTRIFUGAL FORCE - POUNDS.
COEFFICIENTS FOR BEARINGS.

Plate VIII.

U.S.S. MONTEREY.
(HIGH LEADING)

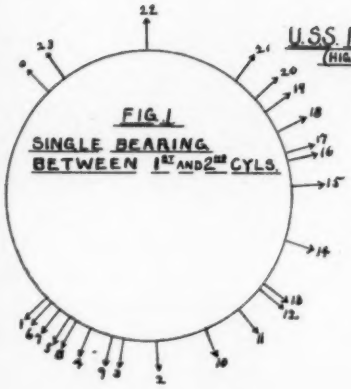


FIG. 1.
SINGLE BEARING
BETWEEN 1ST AND 2ND CYLS.

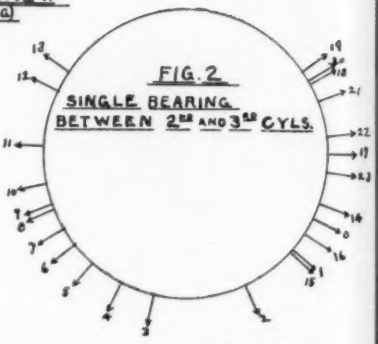


FIG. 2.
SINGLE BEARING
BETWEEN 2ND AND 3RD CYLS.

USS. NEW YORK.
(HIGH LEADING)

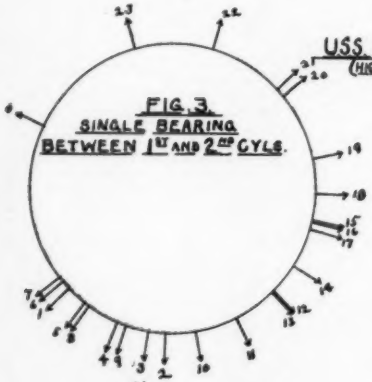


FIG. 3.
SINGLE BEARING
BETWEEN 1ST AND 2ND CYLS.

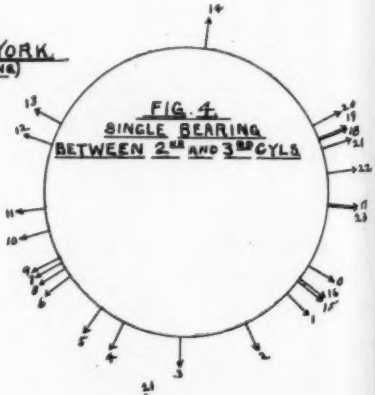


FIG. 4.
SINGLE BEARING
BETWEEN 2ND AND 3RD CYLS.

SS BOSTON.
(LOW LEADING)

FIG. 5.
SINGLE BEARING
BETWEEN 1ST AND 2ND CYLS.

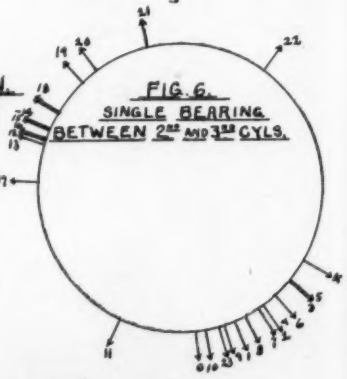


FIG. 6.
SINGLE BEARING
BETWEEN 2ND AND 3RD CYLS.

DIAGRAMS SHOWING DISTRIBUTION OF LOADS UPON A SINGLE BEARING BETWEEN TWO CYLINDERS.

Plate IX.

USS. OLYMPIA.
(LOW LEADING.)

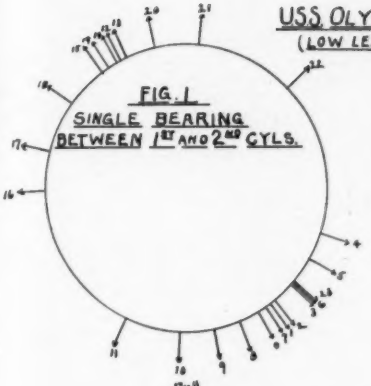


FIG. 1.
SINGLE BEARING
BETWEEN 1ST AND 2ND CYLS.

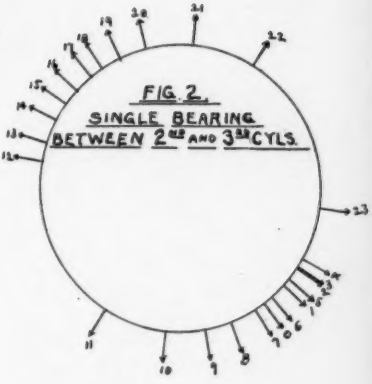


FIG. 2.
SINGLE BEARING
BETWEEN 2ND AND 3RD CYLS.

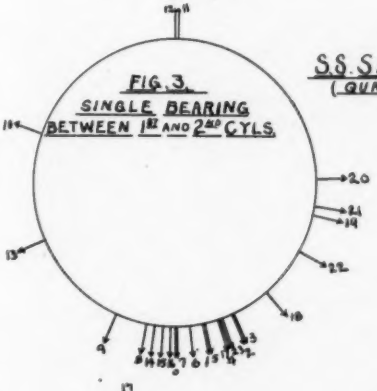


FIG. 3.
SINGLE BEARING
BETWEEN 1ST AND 2ND CYLS.

S.S. SIBERIA.
(QUAD.)

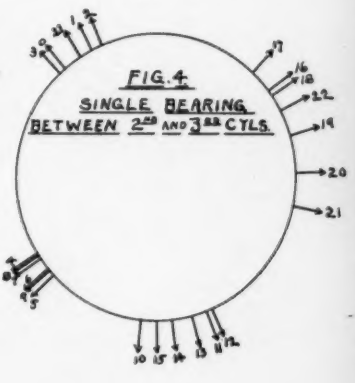


FIG. 4
SINGLE BEARING
BETWEEN 2ND AND 3RD CYLS.

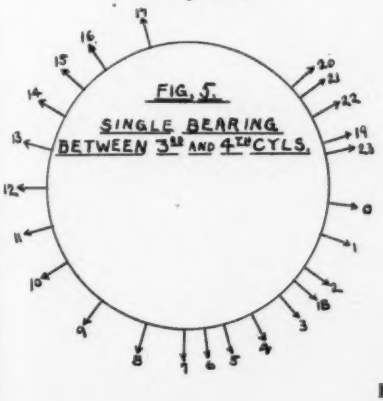


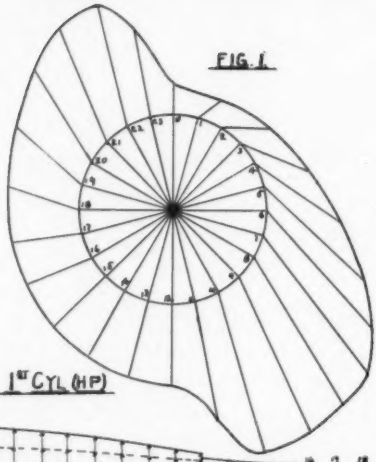
FIG. 5.
SINGLE BEARING
BETWEEN 3RD AND 4TH CYLS.

DIAGRAMS SHOWING THE DISTRIBUTION OF LOADS UPON A SINGLE BEARING BETWEEN TWO CYLINDERS.

Plate X.

XU

FIG. 1



1ST CYL (HP)

FIG. 9

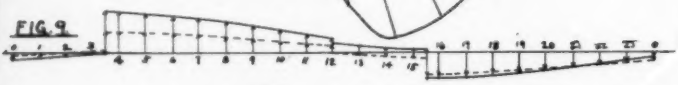


FIG. 5

DISTRIBUTION OF LOADS ON BEARINGS OF 1ST CYL.

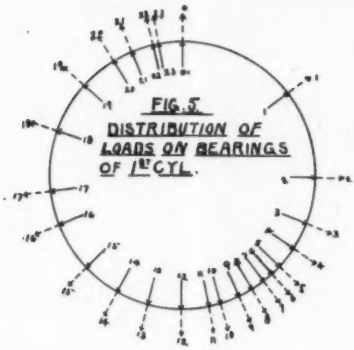


FIG. 6

DISTRIBUTION LOADS ON BEA OF 2ND CYL.

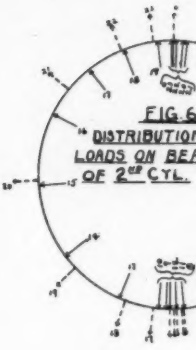
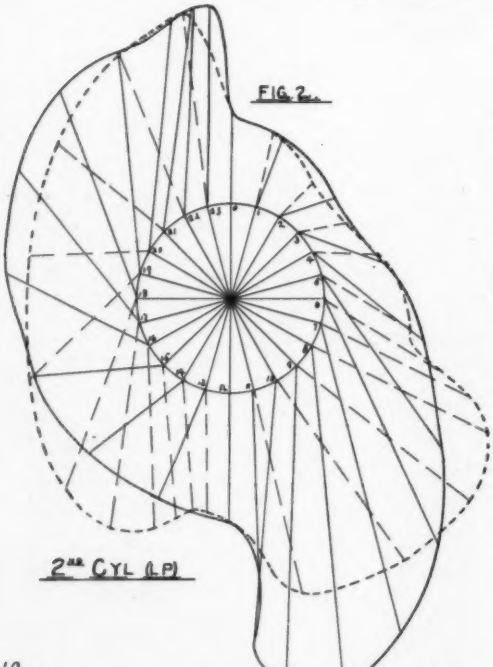
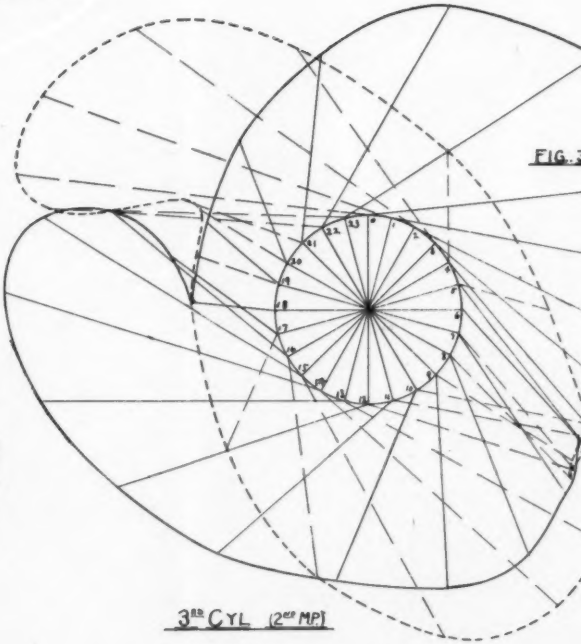


FIG. 2.



2ND CYL (LP)

FIG. 3



3RD CYL (2ND MP)

FIG. 10

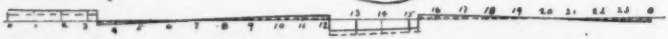


FIG. 11

Plate V

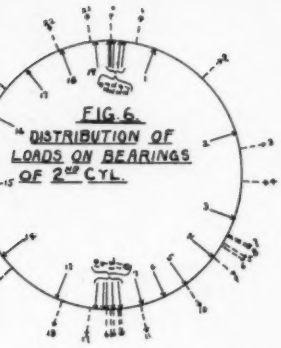


FIG. 6.
DISTRIBUTION OF LOADS ON BEARINGS OF 2ND CYL.

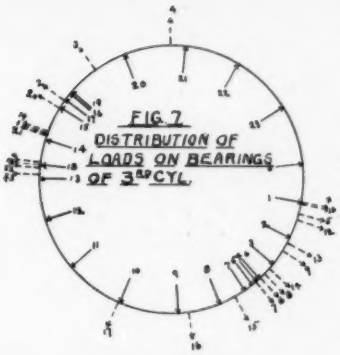


FIG. 7
DISTRIBUTION OF LOADS ON BEARINGS OF 3RD CYL.

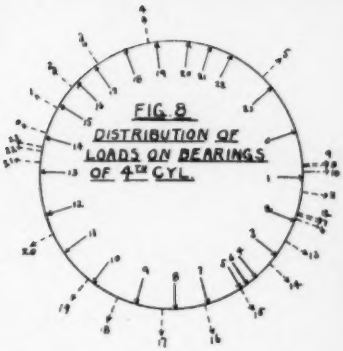


FIG. 8
DISTRIBUTION OF LOADS ON BEARINGS OF 4TH CYL.

DIAGRAMS SHOWING INTENSITY, DIRECTION AND DISTRIBUTION OF LOADS ON THE MAIN BEARINGS OF THE ENGINES OF

S.S. SIBERIA

FULL LINES ARE FOR AFT BEARINGS OF CYLINDERS
BROKEN " " " FOR'D " " "

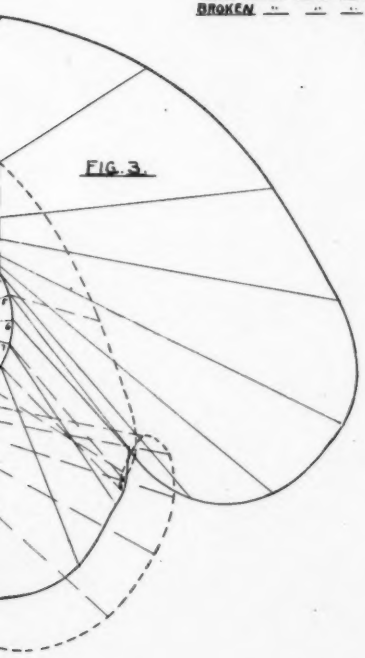
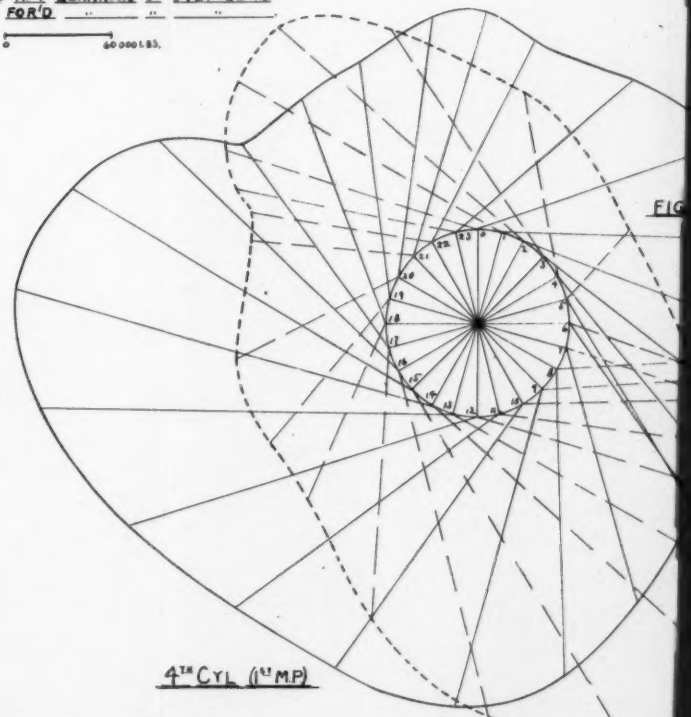


FIG. 3.



FIG

4TH CYL. (1ST M.P.)

FIG. 12.

Plate VI.

FIG. 1

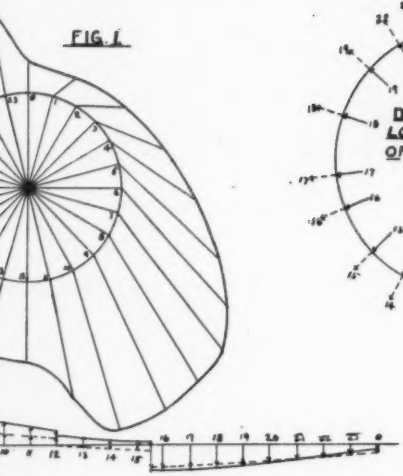


FIG. 5

DISTRIBUTION OF LOADS ON BEARINGS OF 1ST CYL.

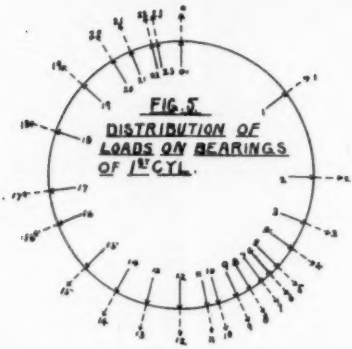
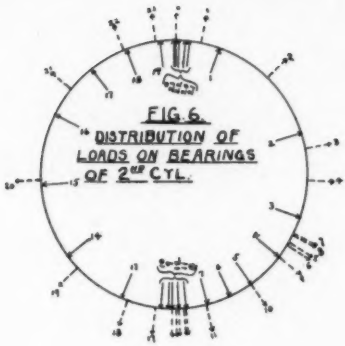
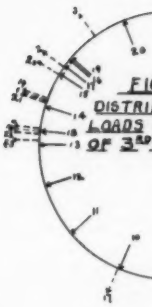


FIG. 6.

DISTRIBUTION OF LOADS ON BEARINGS OF 2ND CYL.





DIAGRAMS SHOWING INTENSIT
OF LOADS ON THE MAIN BEAR
— S.S. SIBER
FULL LINES ARE FOR APT BEAR
BROKEN " " " FOR'D

FIG. 2.

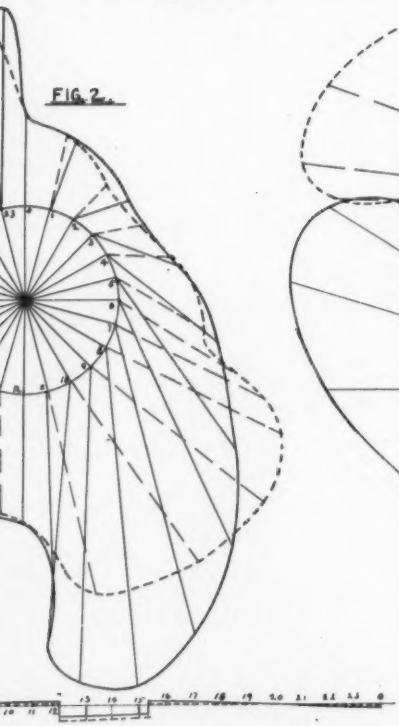
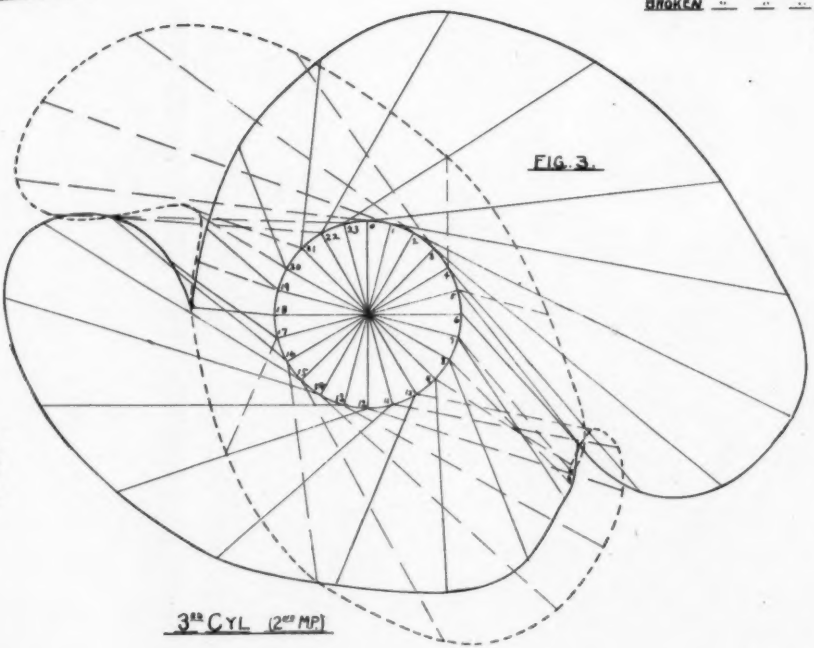


FIG. 3.



3RD CYL (2ND MP)

FIG. 11

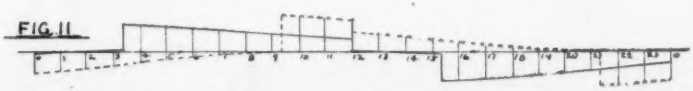


Plate VI.

FIG. 7
DISTRIBUTION OF LOADS ON BEARINGS OF 3RD CYL.

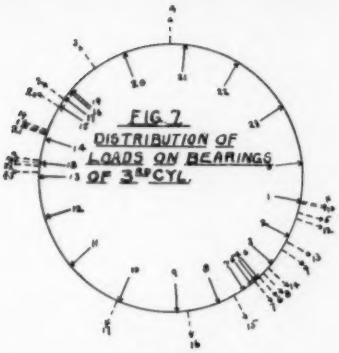
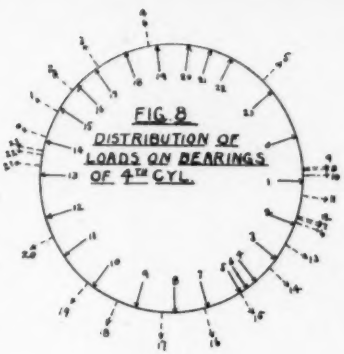


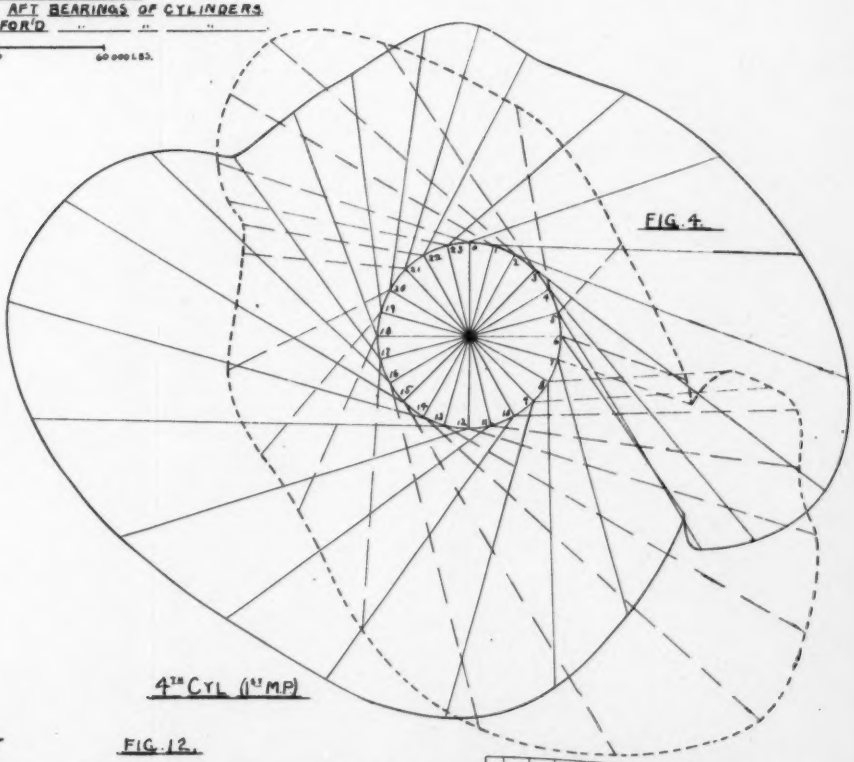
FIG. 8
DISTRIBUTION OF LOADS ON BEARINGS OF 4TH CYL.



RAMS SHOWING INTENSITY, DIRECTION AND DISTRIBUTION
OADS ON THE MAIN BEARINGS OF THE ENGINES OF
— S.S. SIBERIA —
L LINES ARE FOR AFT BEARINGS OF CYLINDERS
KEN " " " FOR'D " " "

60 000 LBS.

FIG. 4.



4TH CYL (1ST MP)

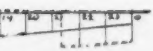
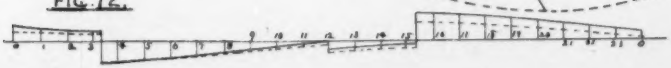


FIG. 12.



THE DETERMINATION OF THE PRINCIPAL DIMENSIONS OF THE STEAM TURBINE WITH SPECIAL REFERENCE TO MARINE WORK.

By Mr. E. M. Speakman.

[Read before the Institution of Engineers and Shipbuilders in Scotland, October 24, 1905.]

In designing any turbine installation, the first and most essential step is to estimate the highest suitable speed of rotation in order that the turbine may be made as small as possible for any given efficiency. While there is now no great difficulty in accurately determining the proportions of turbines and propellers to ensure certain results, with ample confidence to enable a stiff guarantee to be made both for speed and economy, the principal dimensions involve considerable calculation, and much attention must be devoted to the efficiency of propulsion if the best results are to be obtained.

Knowledge of propulsive efficiency, and all its component values for various classes of work, is essential for calculating propeller dimensions, and thereby arriving at the highest revolutions attainable, for it is only when the rotary speed is settled that the best compromise of turbine dimensions, chiefly in the matter of blading arrangements, can be determined.

Turbine efficiency and propeller efficiency must be considered separately and also together, because it may be found that the use of revolutions somewhat below the maximum obtainable will increase the combined efficiency, while on the other hand, to obtain certain advantages in weight and space, this efficiency may be slightly sacrificed at the highest speed, and it is necessary to know the effect of such modifications on the

design and performance. Roughly, the weight of the turbines will vary inversely as the square of the revolutions, while the economy of the turbine will remain almost constant if designed for the same internal conditions; the efficiency of the propeller will be slightly improved as the revolutions decrease and the diameter is made greater.

In electrical work design of the generators almost invariably admits of very convenient speed, but, in marine work, the size of the propeller necessary to avoid cavitation imposes a much lower limit, with which the size of the turbine is greatly increased. The minimum size of propeller required to avoid this phenomenon (and its attendant inefficiency due to increased slip and consequent loss of thrust at the higher powers), *must* be calculated at the beginning of any design, as it is almost impossible to assume certain revolutions, and later on to design a propeller to suit.

Cavitation is partly the result of attempting to obtain too much work per square foot of blade area, and partly of excessive peripheral speeds. It has been found, by bitter experience occasionally, that there is a narrow limit to the tensional pressure possible on the water, per unit of projected area, beyond which the propeller efficiency drops very rapidly. This pressure is approximately from 10 to 12 pounds per square inch at a depth of twelve inches below the surface, and to reduce the total thrust to this, sufficient blade area must be provided, which, in conjunction with certain practical proportions, necessitates a certain size of propeller, thereby limiting the revolutions.

Friction and slip constitute the normal losses in all propellers, and augmented resistance must also be taken into account. This latter loss, however, is materially reduced with the smaller diameters of propeller found in turbine work. Comparing the *Manxman* with the *Ulser*—a Holyhead mail boat of similar speed and power—the total disc area of the twin propellers of the latter is 226 square feet (for two twelve-foot diameter propellers), while that of the *Manxman* is only about 80 square feet. If the thrust deduction is

proportional to the absolute area of the disturbance of the steam lines at the stern, the effect on the *Ulster* will be far greater than on the *Manxman*, but this action is to some extent affected by the intensity over the disturbing area which again is modified by the proximity of the propellers to the side of the vessel, this being less in turbine work. The disc area in H.M.S. *Velox* is less than half that of the propellers in ordinary destroyers of the same power and speed. Cavitation is a preventable loss, and its presence on many vessels with insufficient blade area may be deduced from the falling off of the thrust curve and the rapid rise in the slip curve above a certain speed.

From the analysis of numerous trials it appears that the pressure per square inch of projected area, when reduced to twelve inches immersion of tip, due to the effective thrust, is approximately 1 pound for every 1,000 feet per minute of circumferential velocity of blade tips. For a screw of a given pitch ratio, working at its maximum efficiency, this velocity should be proportional to the designed speed of the ship, and at full speed the pressures seem to have hitherto been about 5 pounds per square inch for slow cargo vessels, from 6 to 7 pounds for ocean-going mail steamers, and from 7.5 to 8.5 for cross-channel steamers; in cruisers and battleships they vary from 8 to 10.5 pounds in some recent notable instances, in a torpedo craft from 9 to 11 pounds. At about 9 pounds, rather less possibly, the lowest suitable limit is reached for turbine screws; from 10 to 11 pounds may be more usual in fast vessels, and though even from 12 to 14 pounds has been known, pressures over about 11 pounds always seem to be accompanied by low efficiencies. In large ocean-going vessels, which may be delayed by head winds and seas, a much lower designing pressure should be used, but, in destroyers, as suggested above, something may be sacrificed at the maximum speed to obtain other advantages. The above pressure is worked out as mean pressure, as there exists no method of determining the local intensity per square inch, though the tendency of the distribution may be assumed in

some cases. The only published results are in Barnaby's papers on the trials of H.M.S. *Daring*, but they are very inconclusive in many ways. Fig. 1 is submitted with much diffidence, merely as an illustration of the values of this limiting pressure.

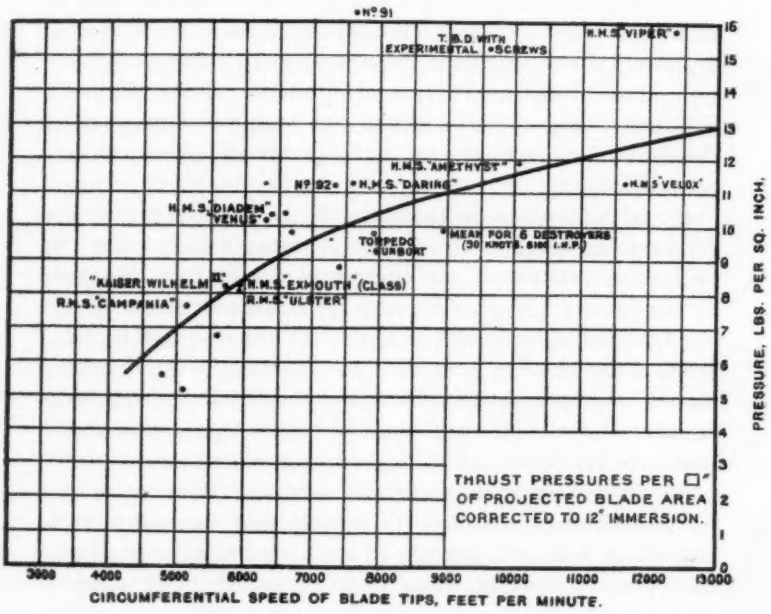


Fig. 1.

The maximum peripheral speed of tip ever used, I think, was 12,400 feet per minute in H.M.S. *Viper*; in H.M.S *Velox* it is about 11,650, and in the *Londonderry* about 11,760, but many vessels have been below 9,000, which is quite normal for ordinary destroyer practice.

The pitch ratio for turbine propellers has been purposely made considerably finer than usual. Thus, the pitch ratio for the *Emerald* was about 0.6; in channel steamers and cruisers of from 18 to 25 knots it has varied from 0.8 to 1.0, and in torpedo craft from 1.0, in H.M.S. *Velox* to 1.35 in

H.M.S. *Viper*, and 1.6 in H.M.S. *Cobra;* the latter vessels having 1, 2 and 3 screws per shaft respectively driven by identical turbines, the approximate revolutions at full speed being 900, 1,200 and 1,050 for 27, 36 and 31 knots respectively on trial.

The percentage of slip has varied from 28 per cent. in H.M.S. *Viper* down to about 14 per cent. in the *Viking*—channel steamers usually having about from 17 to 24 per cent. For large ocean-going vessels, about 16 to 20 per cent. may be used with due regard to other considerations of propeller efficiency. The section of the blades should be carefully designed in order to try to obtain a shape that will enable as high a mean pressure as possible to be adopted. Recent experiments in the model tank at Washington, D. C.,* seem to show that a symmetrical section, as shown in Fig. 2, will materially increase the pressure at which cavitation commences, and also demonstrate that in fine-pitch high-speed screws the back of the blade should receive almost as much attention as the face. This, as the author is well aware, is no new

*See D. W. Taylor's paper, "American Soc'y N. A.," 1904.

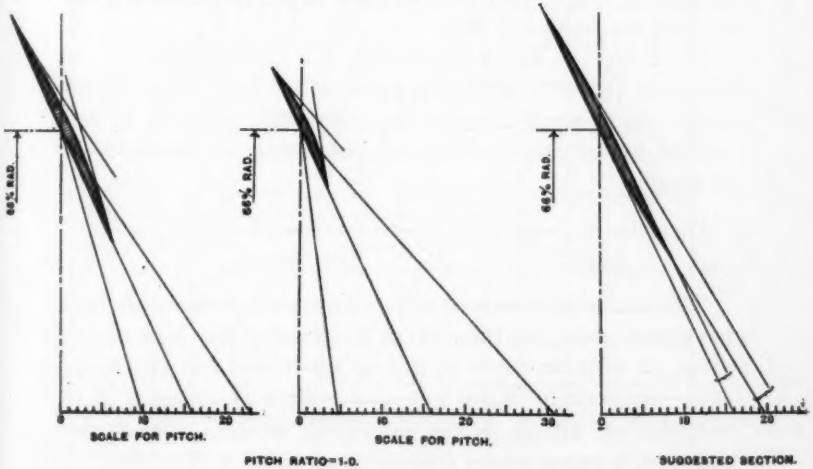


Fig. 2.

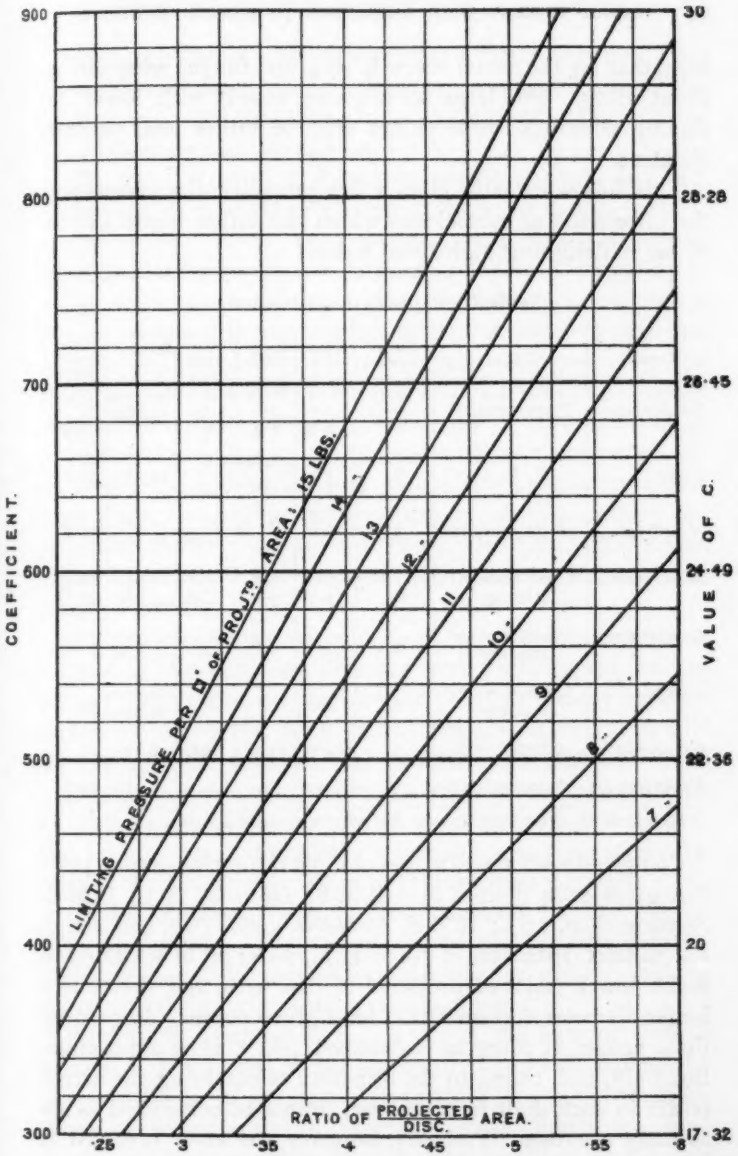
idea, but there have been repeated indications, especially in the trials of ordinary torpedo-boat destroyers, that while the gain may not be great, it is sufficient to merit attention. Mr. Parsons has advocated a 10 per cent. reduction of pitch at the blade tip in order to avoid excessive local thrust, which might induce early cavitation, but there seems to be no advantage from departing from a true screw. The tendency of late years, in reciprocating-engine practice, has been to increase the ratio of projected to disc area from the .2 of Froude's classic screw, and the .22 to .26 of naval practice, to about .33; destroyer practice is included between this and .37, or even .4, at which point turbine practice may be said to commence. In this even from .5 to .56 has been used, but beyond, about .58 blade interference becomes excessive, and to obtain greater area a larger diameter must be used.

The best form of blade is still undetermined. In the photo of the stern of the *Lorena*, Fig. 3 (omitted in this report), will be seen the usual shape adopted, and experience seems to show that this almost circular shape, with the area disposed symmetrically on each side of the center line, and with the generating line of the screw at right angles to the axis, gives as good results as any form.

I find that the following formula will give the diameter of a turbine propeller with considerable accuracy when the effective thrust along the shaft is known, and this must be calculated in any case, if the steam balance of the turbine is to be good:

$$\text{Diameter of propeller in feet} = \sqrt{\frac{\text{effective thrust in lbs.}}{\text{coefficient}}} = \frac{\sqrt{T}}{C}.$$

This coefficient has been deduced from the limiting pressure per square inch, and the ratio of projected to disc area, and is given in diagram form, in Fig. 4, where the full coefficient 400—900 is given in the left-hand scale, and values of C or $\sqrt{\text{Coefficient}}$ appear in the right-hand margin. The square root is only extracted for simplicity, whereby such coefficients as 30 are obtained for H.M.S. *Viper;* while for the *Manx-*



CO-EFFICIENTS FOR TURBINE PROPELLERS.

$$\text{DIA. OF PROPELLER IN FEET} = \sqrt{\frac{\text{ACTUAL THRUST OF SCREW IN LBS.}}{\text{CO-EFFICIENT CORRESPONDING TO DESIRED PRESSURE AND AREA RATIO.}}} = \frac{\sqrt{T}}{C}$$

Fig. 4.

man, that for the center screw is 26.4, and for the wing screws about 28.75. For large ocean-going vessels with lower designing pressures these values will be rather less, perhaps about 22.

This following table gives a few propeller dimensions and the corresponding coefficients, which the author trusts will be of use in designing high-speed screws.

Table I.—PROPELLER DIMENSIONS.

Vessel.	Type.	No. of screws.	Diameter.	Pitch.	Pitch Ratio.	Speed of tip.	*C* Approximate.
			Ft. In.	*Ft. In.*		*Ft.p.min*	
Turbinia.....	Experimental	9	1-6	2-0	1.33	10,860	28 *
		3	2-4	2-4	1.0	...	31.3 *
Viper.....	T. B. D.	8	3-4	4-0 fwd 4-6 aft.	1.2 1.35	12,350	30.1
Amethyst.....	3d class cruiser	$3\frac{1}{2}$	6-6	6- 6 5-10	1.0 .898	9,200 10,000	30.8
Manxman...	Cross-channel steamer	$3\frac{1}{2}$	6-2 5-7	5-7 5-0	.906 .896	10,270 10,760	26.4 28.75
Londonderry	Cross-channel steamer	$3\frac{1}{2}$	5-0	4-6	.9	10,550 11,810	30.8
Dieppe.....	Cross-channel steamer	3	5-3	..	...	10,100 10,400	29
Carmania.....	Atlantic mail	3	14-0	13-0	.928	8,125	21
Victorian.....	Intermediate	3	8-30	...	...	7,150	24.65

*These *C* values are calculated from the same effective thrust in each case.

Compared with above values of *C*, reciprocating-engine practice gives such figures as:—R.M.S. *Lucania*, 16.5; H.M.S. *Diadem* class, 17.5; H.M.S. *Exmouth* class, 19.0, and standard 30-knot destroyers of 6,100 H.P., 20.8; all of which have much lower ratios of projected to disc area, and therefore a larger diameter and smaller *C* for a given power. Regarding these figures, it must be understood that *C* is an approximation only, and, owing to the difficulty of obtaining the actual power in each shaft in every case, cannot be considered as absolutely accurate. Probably, however, the error involved is under 2 per cent. It seems likely, to some at present un-

determinable extent but within narrow limits for each class of vessel, that the propeller efficiency is proportional to the coefficient *C*, and this seems to be borne out by the trials of the *Manxman* and the *Londonderry*.

Effective thrust is a somewhat subtle subject, and our knowledge of propulsive efficiency is by no means what it ought to be. These considerations will undoubtedly be brought into far greater prominence in the near future, and it is by no means improbable that the Admiralty or certain private owners will require some definite standard in this, just as coal or steam consumption is regulated at present. The more propeller efficiency is studied and understood the greater will be the improvement in the design of turbine installations for marine work; the turbine itself is a comparatively secondary consideration, and while at present propeller dimensions for turbine steamers can be quite as closely determined as those for ordinary work, the exact proportions must necessarily largely remain subject to modification from actual experience.

While a general tendency has been very noticeable towards increasing the propeller diameter and reducing the revolutions, there will, of course, be some point, at present undetermined, at which the triple screws used in turbine work will be distinctly less efficient than ordinary twin screws. Very largely, this is the case at present with triple screws driven by piston engines, on account of excessive thrust deduction and interference, but probably before this point is reached the weight of the turbines will have prevented its adoption.

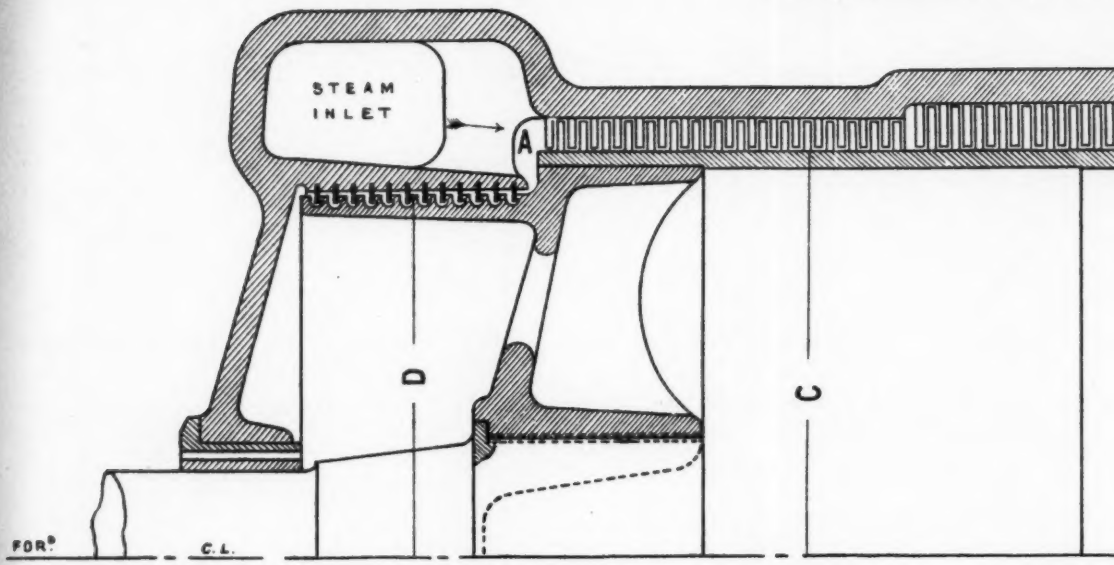
Having obtained the diameter of the propeller and the revolutions possible, the design of the turbine can then be undertaken, but for this no formulæ exist at present, such as are met with in reciprocating-engine practice.

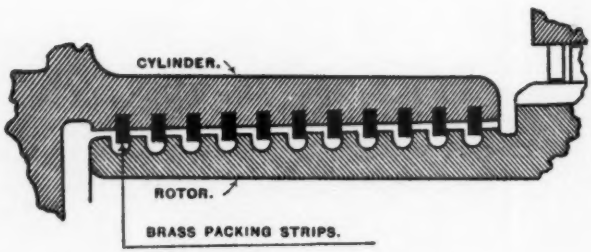
Restricting attention to the design of the Parsons type of turbine, a few notes on the action of the steam among the blades may be of interest. Expanding through a definite range of temperature and pressure, steam exerts the same energy, whether it issues from a suitable orifice or expands

against a receding piston. Two transformations of energy take place in the steam turbine—first, from thermal to kinetic energy ; secondly, from kinetic energy to useful work. The latter alone presents an analogy to the hydraulic turbine, the radical difference between the two lying in the low density of steam compared with water, and the wide variation of its volume under different temperatures and pressures.

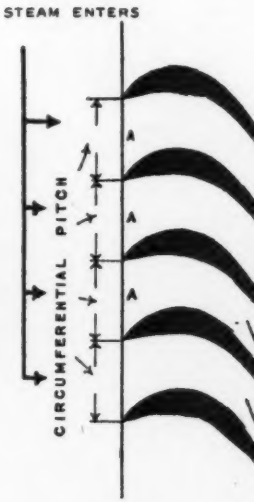
Fig. 5 gives a sectional elevation of a marine turbine blading arrangement, and though this is only for an H.P. cylinder the principle is exactly the same throughout. The expansion, which is approximately adiabatic is carried out in this annular chamber from A to B, which essentially resembles a simple divergent steam nozzle, but with this difference, that whereas in a nozzle the heat energy of the working steam is expended upon itself in producing high velocities, in Parsons' turbine the total expansion is subdivided into a number of steps, in each of which a certain dynamic relationship between jet and vane is maintained. The expansion of steam at any one stage is typical of its working throughout the turbine. Each stage consists of a ring of stationary blades which give direction and velocity to the steam, and a ring of moving blades that immediately convert the energy of velocity into useful torque. The total torque on the shaft is due to the impluse of steam entering the moving blades and to reaction as it leave them, this process being repeated throughout the turbine.

Leakage past the revolving portion of the spindle at D is almost entirely prevented by the ingenious form of frictionless packing, shown on a larger scale in Fig. 6. The fine clearances and the sudden increase of section have the effect of alternately wire-drawing and expanding the steam, so that at successive grooves it becomes increasingly difficult for the steam to leak past the fine clearances. In the astern turbines, a radial form of packing, depending on fine tip clearances, must be adopted owing to the difference in expansion between spindle and cylinder. Numerous varieties of these forms of packing exist, some of them being extremely efficient in their action.





ENLARGED SECTION OF DUMMY PACKING.

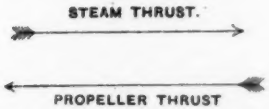


IMPULSE ACTION.

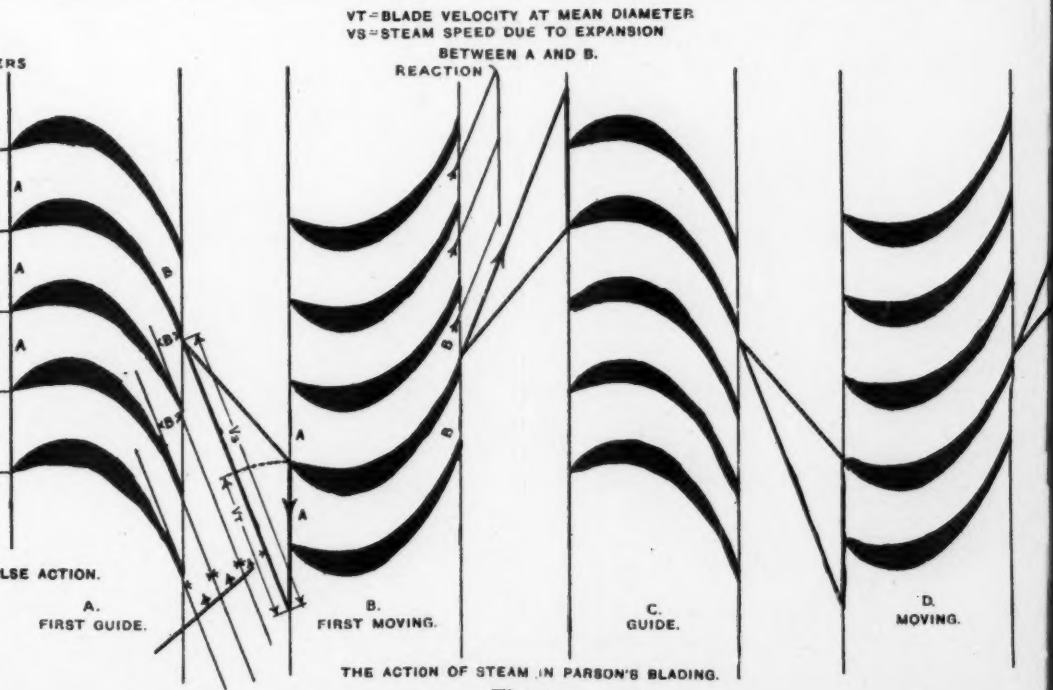
A.
FIRST GUIDE

Fig. 6.

Fig. 5.



SECTIONAL ELEVATION OF H.P. CYLINDER OF A PARSON'S TYPE
OF MARINE TURBINE.



THE ACTION OF STEAM IN PARSON'S BLADING.

Fig. 7.

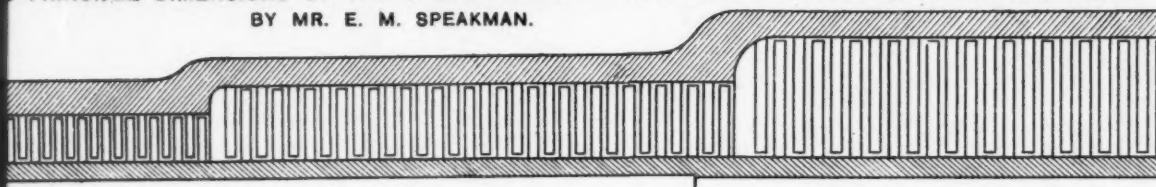
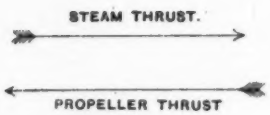
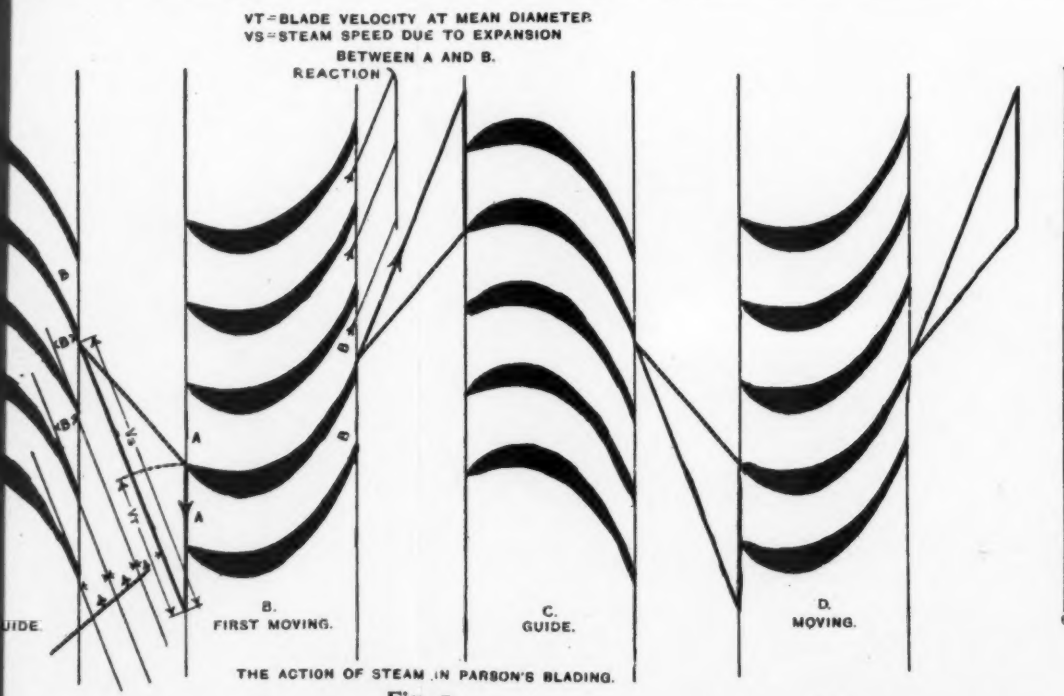


Fig. 5.

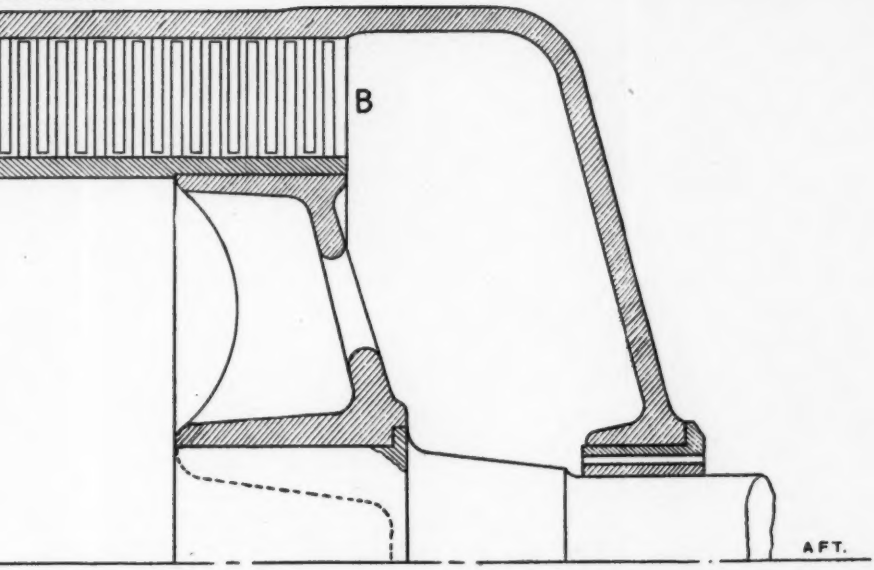


SECTIONAL ELEVATION OF H.P. CYLINDER OF A PARSON'S TYPE OF MARINE TURBINE.



THE ACTION OF STEAM IN PARSON'S BLADING.

Fig. 7.


B
AFT.

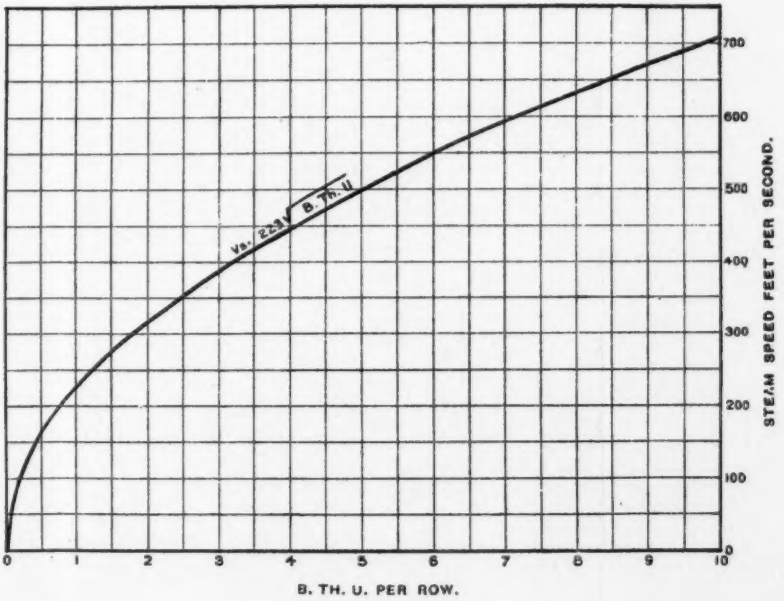

Vs. = 22.3 √ B. Th. U.
STEAM SPEED FEET PER SECOND.
700
600
500
400
300
200
100
0
0
1
2
3
4
5
6
7
8
9
10
B. TH. U. PER ROW.

Fig. 8.

The laws governing the best theoretical velocity of steam and blades are similar to those for water turbines, but in practice some modification is necessary, and the best ratio of blade speed and steam speed is still a matter of opinion. The ideal condition for impulse turbines occurs when the peripheral velocity of the buckets is one-half that of the jet, or in reaction turbines when it is equal to it.

Parsons' turbines, have been built with Vt, Fig. 7, varying from .25 to .85 of Vs, where Vt represents blade velocity at mean diameter, and Vs the steam speed due to expansion across the row in question. A very usual ratio in electrical work for large units has been $\frac{Vt}{Vs} = 0.6$, but this involves a greater number of rows than is possible in marine work, and the ratio must be reduced. These ratios need very careful calculation. The steam consumption must be accurately known in order to proportion them correctly throughout the turbine, and the necessity (which is inevitable with the present form of caulking piece,) of having the same area of openings in so many rows while the steam volume increases so rapidly, adds to the difficulty of close calculation. The potential energy of the steam, corresponding to the "head" in water turbines can easily be calculated for given pressure differences.

B. T. U. $\times$ 778 = Energy in foot pounds per pound of steam $= e$

$$Vs = 8\sqrt{e} = 223\sqrt{\text{B. T. U.}}$$

For a given blade velocity, it is obvious, then, that the speed ratio between jet and vane must affect the number of stages, and the greater the ratio of Vt to Vs the greater will be the required number of rows; that is, to obtain the required Vs at each stage a smaller pressure drop per row is necessary, or *vice versa.*

The best blading arrangement, scientifically and commercially, is the result of much theory and practice. The mean diameter is an arbitrary dimension capable of wide variation

without affecting the efficiency, provided that the number of rows is correct; it is found by assuming, from experience, a blade velocity, whence—

$$\text{Mean diameter in inches} = \frac{\text{Blade velocity in feet per sec.} \times 228}{\text{R.P.M.}}.$$

To arrive at the corresponding number of rows, the revolutions being given, the ratio of Vt to Vs must be settled, from which the steam speed can be obtained; it is a convenient assumption at the beginning of any design to consider the turbine as parallel throughout and of constant efficiency, and to design on this basis. The number of rows N on one diameter can be found by working out the B.T.U necessary to give a certain steam speed at each row, see Fig. 8; the available energy divided by the energy it is desired to abstract at each row will give the number of rows required. This result may be arrived at by various ways, but the principle involved is the same in each case. Numerous empirical coefficients for approximating steam speeds and the corresponding number of rows are obtainable from experience, and are similar in use and value to the Admiralty coefficient; that is, while they represent a crude method of doing something that should be done more scientifically, they are very simple and capable of rapid handling. Being, however, based on long and costly experiments, much reticence is reserved regarding their publication. Varying, of course, with the steam pressure and vacuum, the number of rows on one diameter would involve an excessive length of turbine and also inconvenient blade heights. It is, therefore, usual to divide the rotor into three or more stages, which have the advantage of shortening the turbine and reducing the number of rows. If n = the fraction of power developed in the first cylinder or barrel, $\frac{N}{n}$ = number of rows in the first barrel, and with the alteration of diameter and increase of blade velocity in the succeeding stages, the number of rows on other barrels are so altered as to keep, for equal powers and efficiencies:

$$(\text{Blade velocity})^2 \times \text{No. of rows} = \text{Constant}.$$

The vane speeds adopted in practice vary considerably; for some time 100 feet per second was regarded as a standard for the first row, and I think the Westinghouse Co. at Pittsburg was first to make a radical departure in this and adopt far higher speeds. The maximum vane speed used for Parsons blading is, as far as the author is aware, about 375 feet per second in the low-pressure blades, and 170 in the H.P. blades of electrical turbines; the lowest speeds used are

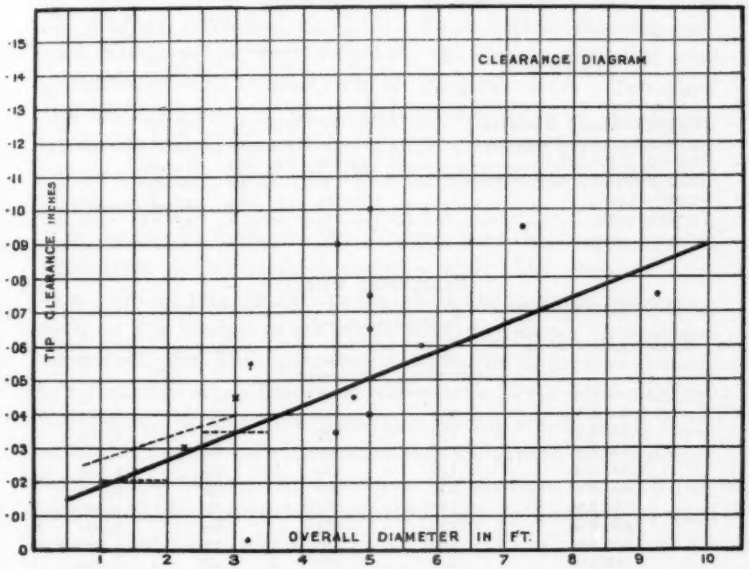


Fig. 9.

in marine work, and are only about one-third of these. To some extent blade speed is governed by blade height; the speed should be so modified that this may be at least three per cent. of the mean diameter to reduce the proportion of clearance losses. Leakage over the tips of the blades is, perhaps, not so detrimental on account of actual leakage loss as in its superheating effect on steam between the row past which it leaks and the last row, because this reheating

effect upsets calculations regarding openings by increasing the steam volume, and thereby affects the fluid efficiency. This leakage over the tips must be taken into account in designing reaction turbines. Temperature and diameter influence the clearance, and the stiffer the cylinder is to resist distortion due to heat the less it may be made. A clearance diagram based on measurements off a large number of machines is given in Fig. 9.

Table II.—MARINE WORK.

Type of vessel.	Peripheral vane speed.		Mean Ratio of $V_1 - V_2$.	Number of shafts.
	H.P.	L.P.		
High-speed mail steamers	70–80	110–130	.45–.5	4
Intermediate mail steamers	80–90	110–135	.47–.5	3 or 4
Channel steamers	90–105	120–150	.37–.47	3
Battleships and large cruisers	85–100	115–135	.48–.52	4
Small cruisers	105–120	130–160	.47–.5	3 or 4
Torpedo craft	110–130	160–210	.47–.51	3 or 4

ELECTRICAL WORK.

Normal output of turbine.	Peripheral vane speed.		No. of rows.	Revolutions per minute.
	First expansion.	Last expansion.		
5,000 kw.	135	330	70	750
3,500 kw.	138	280	75	1,200
2,500 kw.	125	300	84	1,360
1,500 kw.	125	360	72	1,500
1,000 kw.	125	250	80	1,800
750 kw.	125	260	77	2,000
500 kw.	120	285	60	3,000
250 kw.	100	210	72	3,000
75 kw.	100	200	48	4,000

In Table II, the vane speeds adopted in various classes of work are given, and the reduction in peripheral speed on account of the propeller reducing the revolutions, and the necessary proportion of blade height modifying the diameter, may be clearly seen. To this combined action is due the fact that only in the faster classes of vessels, or in those small types in which some propulsive efficiency can be sacrificed, is the turbine applicable. In slow cargo steamers, though the

revolutions may be high enough, the power required is not sufficient to enable a reasonable blade height to be adopted, and it is this consideration, viz: proportion of leakage over blade tips, that curtails the wider adoption of this type of turbine. For the same low peripheral blade speed other types of turbine are unsuitable on account of the impossibility of reducing the steam velocity sufficiently without abnormal weight and inefficiency.

The smallest size of marine turbine is usually larger than the average electrical turbine as far as power is concerned, and therefore does not meet with the same commerical considerations as the smaller sizes of the latter type. These are not designed for the same internal efficiency as the larger machines, chiefly on account of manufacturing cost, and they do not attain anything like the same efficiency compared with the Rankine cycle.

Speaking in reply to the discussion on his paper to the Institution of Naval Architects in 1903, Mr. Parsons said that, "for all practical purposes, while the steam is traversing each set (of blades) as shown, it behaves like an incompressible fluid, just like water would do, as the expansion is very small at each set. The frictional losses and the eddy-making losses would be practically identical within small limits with what they would be with water, and the actual forces would be in proportion to the density of the medium. . . . In the turbine blades themselves, the efficiency is between 70 and 80 per cent."

Using this hydraulic analogy enables us to calculate the number of stages required in a different manner: the "equivalent head," due to the steam pressure, may be found, together with that at each row necessary to give the required velocity, from which both the number of stages and the coefficient of expansion at each stage may be worked out.

In the early marine designs, such as the *Queen Alexandra* and H.M.S. *Amethyst*, the turbine drums were all made of the same diameter, and the higher speed necessary on the L.P.'s was got by running at considerably higher revolutions

than on the H.P. shaft; but, following up the increase in propeller efficiency found to be due to the use of larger screws, the speed for each shaft is now more nearly equal, while the wing drums are made larger in diameter. The vagaries of the following wake, however, necessitate slightly different propeller dimensions on each shaft, or else slightly different revolutions with the same screws; and it is noticeable that in a triple-screw arrangement, the center screw being right-handed and the wing screws revolving outwards, that the starboard propeller is influenced by the center one, and almost invariably revolves at a lower speed. In a four-shaft design, due to the varying wake values at different speeds, and possibly, also, to some unequal distribution of power, the outer screws run slower at low speeds and faster at high speeds than the two inner shafts, but exact data as to this and the possibility of allowing for it in the design, are still wanting.

In all types of turbines—Parsons', Rateau's, Curtis', &c.—a certain ratio must be maintained between the blade velocity and steam velocity, and as steam acquires very high velocities by expansion, the blade velocity must be maintained either by the revolutions or by large diameters, or both. As the weight increases very rapidly with the diameter, and extraordinarily so with the reduction in rotative speed, it is preferable to increase, if possible, the revolutions or the number of stages rather than the diameter, and especially should this be done in cases where, as in the Rateau or Zoelly types, the weight increases more rapidly in inverse proportion to the R.P.M. and the diameter than it does with other types. To increase the revolutions it may be necessary to increase the number of shafts and propellers, thus reducing the power per shaft and the effective thrust through each screw. Increasing the diameter of the turbine adds largely to the constructional difficulties, especially of the cylinder.

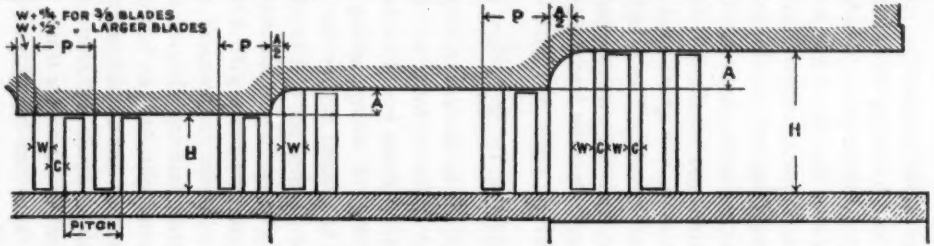
Having obtained the number of rows and the diameter, the blading arrangement can be worked out in detail. The height of blade depends on the volume of the steam and the

speed at which it is to flow, and also on the ratio of the area of exit openings between the blades to that of the annulus between spindle and cylinder, which is about one-third in normal blades. The necessary clear area to pass the steam being equal to volume ÷ velocity, and knowing this annular factor, say 3, for a ratio of one-third (or 2 for ½, etc.), then

$$\text{Height of blade in inches} = \frac{\text{clear area in square inches} \times 3}{\text{mean circumference in inches}}$$

The ratio of blade height to mean diameter should not be less than 3 per cent. or more than 15 per cent., because in the former the leakage will be excessive, and in the latter the bending moment on the blade becomes too great, and the radial divergence of the blades too much. The width of blade, the shape of section adopted, and the circumferential pitch, are standard considerations, and affect the factor 3 given above. It is not proposed to enlarge upon them in this paper. It may, however, be remarked that for $\frac{Vt}{Vs}$ greater than .6 the usual shape of Parsons' section, as shown in Fig. 5, should be modified to a somewhat different form of blade, with a sharper entrance edge. This section is not to be recommended, as, owing to the necessity of strengthening the blade sufficiently, the metal must be placed nearer the exit edge, thus increasing the angle between the face and the back of the exit edge of the blades, and giving, in fact, an inferior shape of opening compared with that obtainable with a blade section adapted to ratios under .6. If, for the present, it is sufficient to use the blade sections and packing pieces similar to those now adopted so generally, in Table III can be found a list of widths for a given height, and the axial spacing of the rows. While this must be kept down to reduce the length of drum, it must be sufficient to allow for some play in overhauling; and sufficient clearance can be allowed here without affecting the economy. The openings between the blades to allow of the passage of the steam are very important, and must be carefully designed. The actual volume of the steam—not the volume per pound, as found in tables, or

TABLE III.—STANDARD BLADING DIMENSIONS.



Height (H)	1″	2″	3″	4″	6″	8″	10″	12″	15″	18″	21″	24″	30″
Width (W)	3/8″	3/8″	3/8″	1/2″	1/2″	5/8″	5/8″	3/4″	3/4″	1″	1″	1 1/8″	1 1/4″
Pitch (P)	1 1/8″	1 1/8″	1 1/4″	1 5/8″	1 3/4″	2 1/8″	2 1/4″	2 1/2″	2 5/8″	3 1/8″	3 1/4″	3 5/8″	4″
Axial clearance (C) ..	3/16″	3/16″	1/4″	5/16″	3/8″	7/16″	1/2″	1/2″	9/16″	9/16″	5/8″	11/16″	3/4″
Section Number													
Spindle Packing Piece No.													
Cylinder, do.													

Note.—While the above represents general practice, it is obvious that such a table is largely arbitrary.

the volume due to adiabatic expansion, but the exact volume per pound at any point along the turbine—must be determined, in order to arrive at the desired adjustment of velocities. It is extremely doubtful whether the present blading arrangements give the best results; greater accuracy of calculation, and consequently improved pressure distribution and efficiency, seem likely to follow the use of a more mechanical blading construction.

Fig. 10 shows the percentage error involved in using either the dry volume, or that due to adiabatic expansion, compared

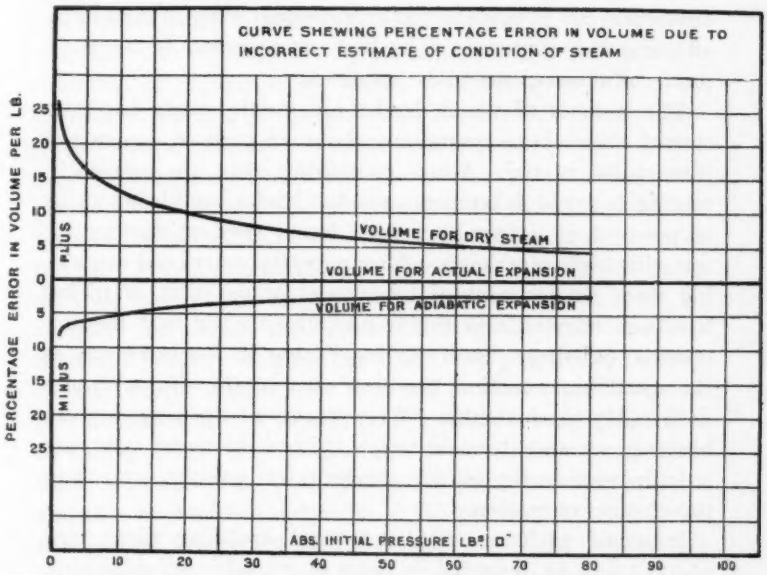


Fig. 10.

with the correct volume corresponding to the actual expansion in a large turbine using dry saturated steam at the first row of blades, and 27 inches vacuum. Attention must be paid to the effect of approximately adiabatic expansion and the consequent moisture in the steam.

For manufacturing convenience, as well as to allow for the expansion of the steam, the blade heights are stepped up, but no rule exists for this; the blades might be of one, or of sixty-four heights, provided the blade openings are correct. It is, however, convenient to step them, as in Fig. 5, say in 8 steps of 8 each, or 4 of 16—even 9 of 7 would do—and so avoid any great variation from the annular area factor 3 as shown above. To obtain heights and areas, it is best to plot off graphically, volumes, steam speeds, and clear areas required. The use of standard blade heights will then enable the number of stages and rows per stage to be determined: wide differences can be made in any arrangement without materially affecting the economy. The best arrangement is largely a matter of convenience and experience.

The material of which blades are usually made is a mixture of cheap brass containing about 16 parts of copper and three parts of tin. Alloys containing zinc are extremely unreliable for high temperatures, but blades containing about 98 per cent. of copper have been found very satisfactory for use with high superheats. More recently a material containing about 80 per cent. of copper and 20 per cent. of nickel has been adopted, and this is undoubtedly the best blading material existing. Steel blading, drawn in the same way as the usual brass section, has been used in the United States with fairly good results. The process of drawing turbine blades gives an extremely tough skin to the metal used, not only increasing the tensile strength, but greatly decreasing the chances of erosion.

It seems probable that the usual caulking piece now adopted will be discarded in favor of a machine-divided strip into which the blades may be fitted, and instead of the slotting, wiring, lacing and soldering process at the tip a similarly machine-divided shroud will be used, giving a far stronger construction, and enabling finer clearances and better workmanship to be obtained at the same time considerably reducing the cost of manufacture and the risk of blade stripping.

The chief causes of the latter may be set down to bad workmanship in fixing the blades, defective blade material, excessive cylinder distortion (this is probably the most fruitful cause, and is a serious one, being due to bad design), whipping of turbine spindles (which is also due to bad design or bad balancing), wear of bearings (which is very remote), and the introduction of extraneous substances such as water or grit. In fact, blade stripping may be said to generally occur from preventable causes. Small vibrations of very high frequency occasionally set up an action in certain rows of responsive length that fatigues the blade material and causes the loss of blades without any fouling at all.*

Due to the action of the steam, an end thrust occurs in the direction of the propeller, which is advantageously used in partially balancing the propeller thrust, thereby reducing the size of thrust block necessary. A margin must be allowed here, and the propeller thrust is not entirely balanced by the pressure on the annulus between the dummy-ring diameter D, and the spindle C, Fig. 5, plus the end pressure on the blades. For the diameter D to give the required annulus, as well as that of the propeller, the effective thrust must be carefully calculated ; and experience shows that there is a drop in steam pressure varying from 10 to 15 pounds per square inch between the pipe inlet to the H.P. receiver and the first row of blades, which should be considered in designing this balancing area. The number of rows of dummy packing used varies according to the designer's judgment very largely, and may be modified according to the pressure and the clearance allowed—say a 7-1000th to a 15-1000th of an inch in electrical work, and rather more in marine work.

The dimensions of the astern turbine are arrived at in the same manner as those of the ahead, the efficiency being largely sacrificed on account of weight and space ; generally the mean diameter is made practically the same as that of the H.P. drum.

* The writer had experience of this early in 1905, when a 5,000 kw. turbine, under test in one of the large New York power stations, shed several rows of blades. This difficulty has also occurred in Europe, and can be circumvented by an alteration in the position in the lacing strip.

To a large extent, the inferior maneuvering capabilities of the earlier turbine steamers were due to insufficient astern power.

It may be remembered that in a marine turbine the spindle is in compression and the cylinder in tension when working.

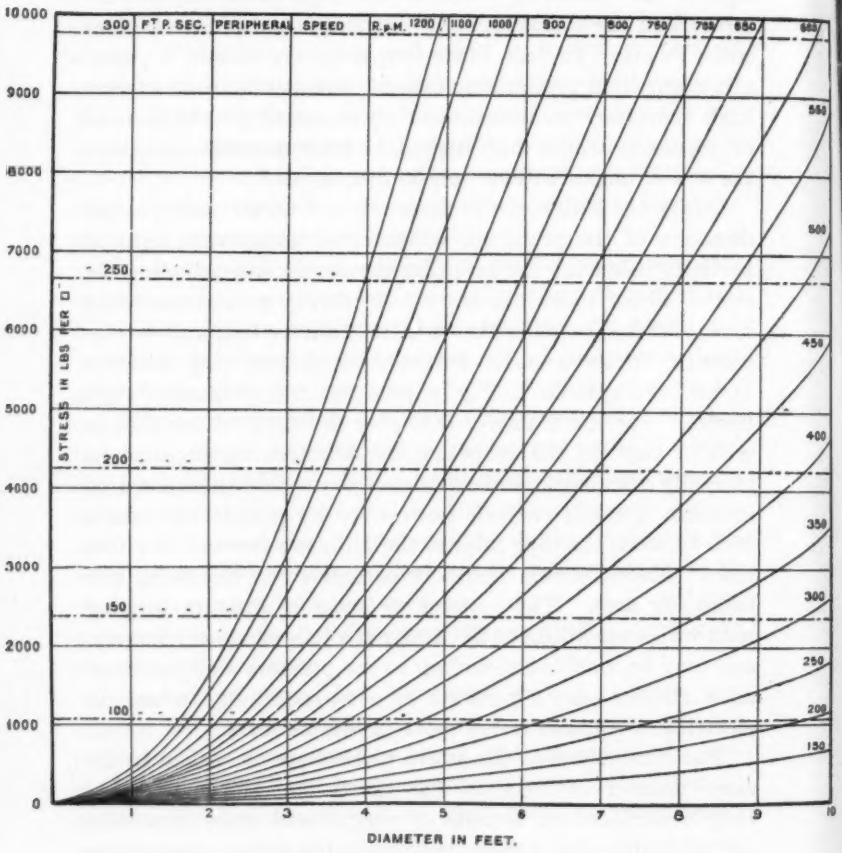


Fig. 11.

In electrical turbines, where the end thrust must be eliminated by the use of balancing pistons, the spindle is in ten-

sion and the cylinder is balanced. The shafts between the turbine bearings and the drum must be made amply stiff enough, as well as strong enough, for any sag in the spindle will destroy the clearance. As will be seen from Fig. 11, the stresses due to centrifugal force are very low in the Parsons turbine, and, except in occasional L.P. barrels, do not exceed about 7,500 pounds per square inch, while at the H.P. end they are usually under 2,000.

The pressure on the bearings in a turbine is only due to the weight of the spindle, plus the negligible addition in marine work, of that due to any gyroscopic actions; it may be taken as from 80 to 90 pounds per square inch as long as the rubbing velocity does not exceed 30 feet per second. If it does, the pressure must be reduced so that the product of pressure $\times$ velocity does not exceed 2,500—2,700. In land work, 50 pounds $\times$ 50 feet is very common. The friction heat of the bearings added to that due to conduction through the pedestals necessitates the use of large oil coolers, and in the case of very high temperatures, of special kinds of oil. If possible, the bearing temperature should not exceed from 140 degrees to 150 degrees F., though the writer has known of 190 degrees F. being used without trouble. In marine turbines this temperature is usually much lower. Rigid bearings are used for marine spindles, not the flexible type adopted in land work.

Space does not permit of more than passing reference to cylinders; but it would be difficult to exaggerate the importance of very careful design in this connection. Cylinders, with heavy flanges on the center line, distort in a very curious fashion when heated with their axis horizontal, and measurements taken off a hot cylinder on a surface plate with micrometer gauges reveal some very remarkable facts. When working, the temperature along the cylinder falls possibly from 400 degrees to 100 degrees F. in a distance of 6 or 8 feet, and, unlike the reciprocating engine, this remains constant; the radial expansion is consequently more at one end than the other; while at any point along the turbine the tendency is

to expand less at the flanges than at the top and bottom. For this reason ample clearance must be allowed; exactly what this will be when spindle and cylinder are hot is hard to say, but it seems most likely that the total clearance area will differ but little from what it is when cold.

The longitudinal expansion when hot is often very marked, and in all turbines necessitates provision for the resultant movement at one end. In marine work the after end of the cylinder is secured to the vessel, the engine seating also performing the function of a thrust-block seat, while the forward end slides forward, taking with it the entire shafting. The thrust block is at the forward end of the cylinder, and also performs the duties of an adjustment block for setting the longitudinal clearances, to do which generally necessitates uncoupling the shafting abaft the turbine.

The difference in expansion between the cylinder and spindle, from the thrust block to the dummy ring, may be the cause of serious difficulties in large marine turbines, unless the closest attention is paid to this feature in the design; and warming up with these large cylinders needs possibly even more care than is essential with large piston engines.

On shipboard, the turbine cylinders are practically under one's feet, and the radiation from them is very unpleasant, especially if there is any leakage from the glands. To all who are responsible for the lagging of cylinders and the system of ventilation in turbine-engine rooms I would call attention to the possibility of their having to stand a watch of from 4 to 6 hours on the top of the H.P. cylinder, such as is the case in the *Eden* or *Amethyst*, the heat in the latter vessel being almost unbearable. With reciprocating engines, one stands on a comparatively cool lower platform with the cylinders overhead, and with some chance of the hot gases rising clear, but in naval turbine work, under a low deck, this point has not met with adequate attention.

In the course of operation, more especially in marine work where no superheaters are used, there is a distinct tendency

for the turbine to be supplied with wet steam, the effect of which on the economy is very marked. Experiments that have been made, show that the percentage increase in consumption is about twice that of the moisture in the steam. For instance, with 2 per cent. of moisture in the steam at the first row, the consumption is increased about 4 per cent.

A considerable amount of data on the performance of turbines compared with reciprocating engines for marine work, is now available. The Admiralty has had tested both cruisers and torpedo-boat destroyers, exactly similar but for their engines and propellers, and trials of the Midland Railway Company's steamers and other cross-channel boats have corroborated the results regarding economy obtained from the naval vessels. In Fig. 12 is given the steam consumption per unit of power of H.M.S. *Amethyst* * compared with that of several recent warships, and it is noticeable that only below from 55 to 60 per cent. of their full speed does the consumption of the turbine exceed that of the piston engines. Very seldom do vessels steam below these speeds. Cruisers carrying relief crews to the China or Australian stations usually proceed at about 60 per cent. of full speed, and in the Atlantic maneuvers of 1903 nearly 80 per cent. of full speed was maintained by the large fleets, whilst the Japanese battleships built in England made their first voyage to Japan at about 54 per cent. of their full speed; at which ratio the consumption per I.H.P. of both the *Hindustan* and *Dominion*, representing very recent battleship construction by eminent builders, is materially in excess of that of the first installation of warship turbines (not including the destroyers). The total consumption of H.M.S.'s *Amethyst* and *Topaze* plotted on a base of power is given in Fig. 13, while Fig. 14 shows that for the Midland Railway boats.† The progressive trials of H.M.S. *Amethyst* are shown in Fig. 15, and, in view of the

* Since the above-mentioned results were obtained, the steam piping has been altered so as to permit the auxiliary exhaust steam to pass through the main L.P. turbines when desired. This arrangement considerably decreases the consumption at low speeds, bringing the *Amethyst's* consumption below that of her sister ships down to ten knots, or about 45 per cent. of full speed.

† Fig. 14 is compiled from the results given in Mr. Gray's paper to the Institution of Naval Architects, July 1905.

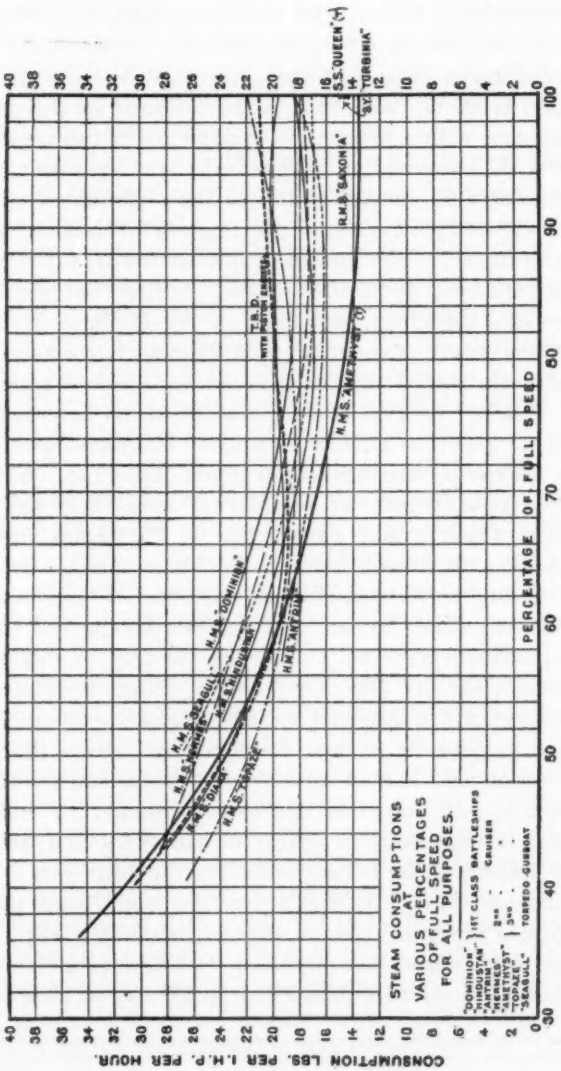

CONSUMPTION LBS. PER I.H.P. PER HOUR.
PERCENTAGE OF FULL SPEED
STEAM CONSUMPTIONS AT VARIOUS PERCENTAGES OF FULL SPEED FOR ALL PURPOSES.
"DOMINION" } 1ST CLASS BATTLESHIPS
"HINDUSTAN" }
"HERMES" 2ND " CRUISER
"HIGHFLYER" } 3RD " "
"TOPAZE" }
"SEAGULL" TORPEDO GUNBOAT
H.M.S. "DOMINION"
H.M.S. "SEAGULL"
H.M.S. "HERMES"
H.M.S. "HIGHFLYER"
H.M.S. "TOPAZE"
H.M.S. "ANTRIM"
H.M.S. "SPARROW"
H.M.S. "SPEEDY"
T.B.D. "ERICSSON"
WITH TURBINE ENGINES
H.M.S. "ANTRIM" (2)
R.M.S. "SAXONIA"
S.S. "QUEEN" (1)
S.S. "TURBINIA"

Fig. 12.

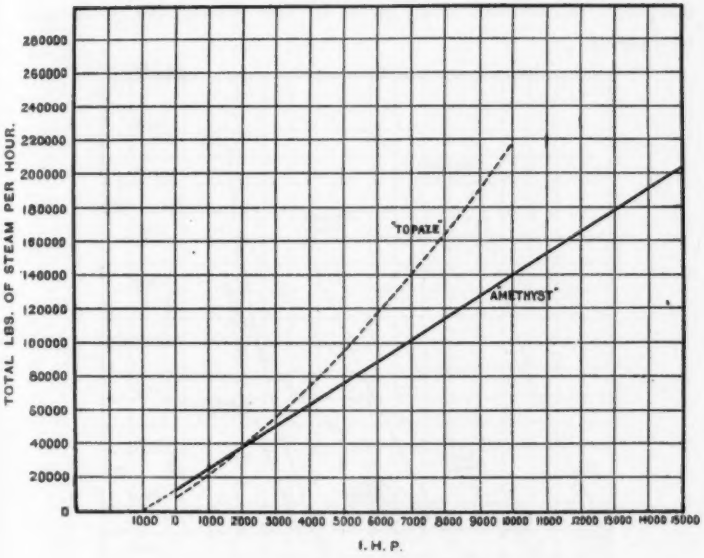

TOTAL LBS. OF STEAM PER HOUR.
280000
260000
240000
220000
200000
180000
160000
140000
120000
100000
80000
60000
40000
20000
0
"TOPAZE"
"AMETHYST"
1000 0 1000 2000 3000 4000 5000 6000 7000 8000 9000 10000 11000 12000 13000 14000 15000
I. H. P.

Fig. 13.

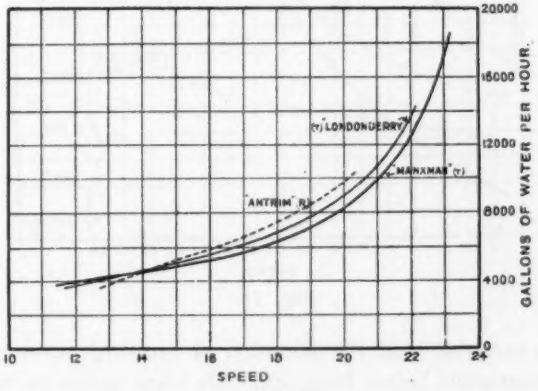

GALLONS OF WATER PER HOUR.
20000
16000
12000
8000
4000
0
(T) LONDONDERRY
MANXMAN (T)
ANTRIM (T)
10 12 14 16 18 20 22 24
SPEED

Fig. 14.

results obtained from these various vessels, the wholesale adoption of turbine machinery in the Royal Navy is not surprising.

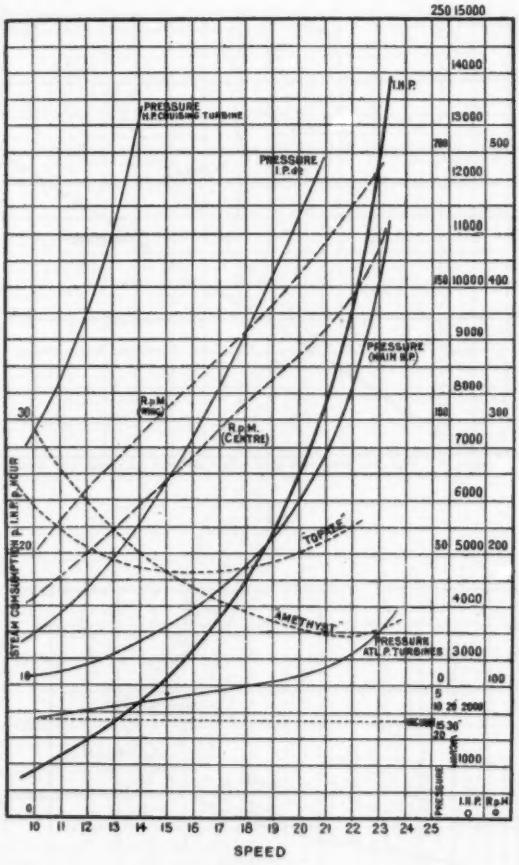


Fig. 15.

It is probable that the adoption of cruising turbines will be discontinued before long, and this view seems to be corroborated by the consumption trials of the Midland Railway steamers. Down to 60 per cent. of her full speed, the *Manx-*

Fig. 16.

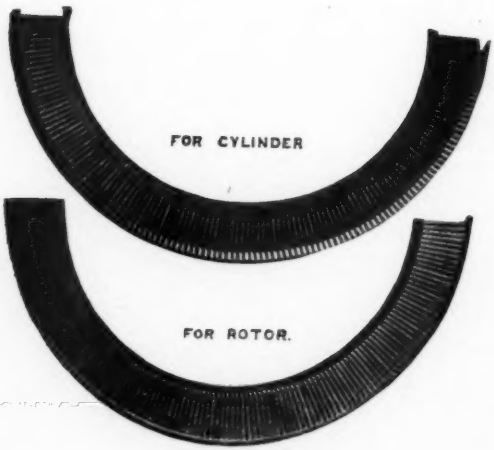


Fig. 17.

man required less water than the highly efficient *Antrim*, and with a different blading arrangement in the main turbines, such a result should be equalled, if not improved on, in war vessels. The additional complication involved with two cruising turbines and their accompanying leakage and receiver losses, together with a considerable increase in weight and space occupied, largely modifies any advantages obtainable in the way of reduced consumption at lower powers. An improved (and easily obtainable) design of main turbine blading should give a better result at the highest powers, practically the same at intermediate powers (as in the case of H.M.S. *Amethyst*, from fourteen to twenty knots), and only slightly inferior at speeds below fourteen knots, while it will undoubtedly admit of greater ease of handling and be much simpler. In a triple-shaft arrangement the unequal distribution of power on the wing shafts, due to the use of cruising turbines, is a distinct disadvantage: the fluctuation in rotative speed, due to shutting off the H.P. cruising turbine, may be seen from the trials of H.M.S. *Amethyst*.

One of the L.P. rotors of the Allan Line steamer *Victorian* is illustrated by Fig. 16. This rotor is 8 feet in diameter, and carries 80 rows of blades, varying successively from 1½ inches to 7 inches in length.

Fig. 17 shows the complete blade rings of a Willan's turbine, the machine-divided construction of which, coupled with the strong form of shrouding adopted, presents such great advantages over the present unmechanical system that its universal adoption may be expected in the near future.

In conclusion, the writer would remark that it is impossible in the scope of a paper such as this, to touch more than lightly on a subject which is of such vast importance. Many of the points dealt with above, such as cavitation, blading, cylinder design, etc., would require a volume to describe. While turbines perhaps are still in their infancy, they are already supplanting the reciprocating engine in many types of vessel. The next few years will undoubtedly show as great an improvement as has taken place since the advent of the *King*

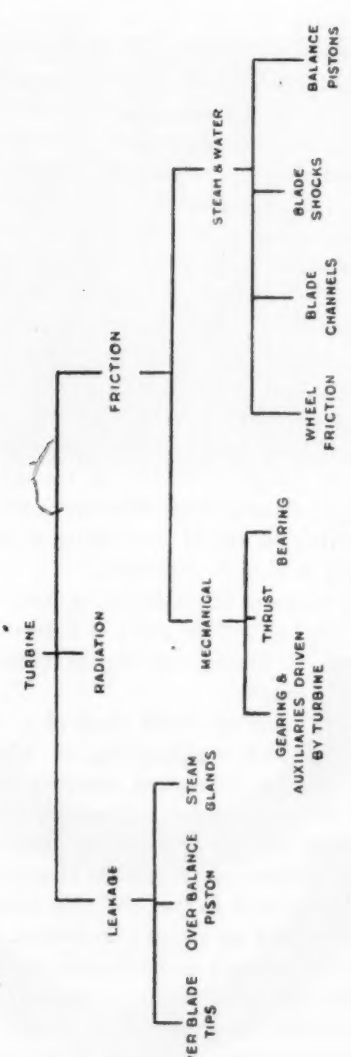

TURBINE
RADIATION
LEAKAGE
OVER BLADE TIPS
OVER BALANCE PISTON
STEAM GLANDS
FRICTION
MECHANICAL
GEARING & AUXILIARIES DRIVEN BY TURBINE
THRUST BEARING
STEAM & WATER
WHEEL FRICTION
BLADE CHANNELS
BLADE SHOCKS
BALANCE PISTONS

Fig. 18.

Edward barely five years ago, the more especially as the subject will, henceforward, be engaging the attention of all engineers instead of a few specialists, and that this improvement will more than justify the policy of the Admiralty and of the Cunard Company is already certain.

It should be a matter of some satisfaction to the members of this Institution that the present status of the marine turbine, if not absolutely due to the Clyde alone, is at any rate entirely a product of Great Britain.

DISCUSSION.

Mr. E. G. Izod (Rugby), speaking as an ex-marine engineer, said he thought it was very evident that the turbine was the only engine of the future for marine work, though no doubt progress would be so rapid that it was probable that the design of a marine turbine installation of ten years hence would very materially differ from that adopted at the present day. Mr. Speakman and himself had worked together on the turbine question, and he could assure the members of the Institution that Mr. Speakman had very closely studied the problems relating to propellers and turbines with a view to obtaining better efficiencies in the propulsion of large and small vessels, and he quite thought that the information contained in the paper, which they had just listened to, would make those interested take a deeper and more careful view of this, one of the most important changes in the annals of marine engineering. He quite agreed with Mr. Speakman's remarks that no information was forthcoming from those who had been building turbines; and while, no doubt, the reticence displayed was justifiable from the point of view that such experimental work was a commercial asset they did not wish to part with, yet one could not help feeling that such reticence was perhaps impairing the chance Britons had of making their turbines, as their reciprocating engines were, second to none in the world. However, he hoped that the extraordinary keenness now displayed by marine engine builders in this new class of work would result in the success which they undoubtedly deserved. Mr. Speakman had called attention to the sample ring of blading exhibited during the evening, and, speaking as a member of the firm which made its blading on this principle, he would like to point out that there were

many advantages which accrued from the use of separately built-up machine-divided blade rings—as for instance, correct spacing of blades, correct angles and openings, increased facility for handling the blading and completing the necessary blading work before the rotor and casing were finished. One of the features of the system was the channel shrouding, which gave extraordinary stiffness to the blade rings, and had the added advantage that if contact did take place between the revolving and fixed elements, no harm was done, as it was practically impossible to disturb the blading even under the severest conditions known at the present time—a factor of great importance in a marine turbine where absolute breakdown of any portion of the propulsive agent was to be avoided. He would, however, point out that this shrouding made a virtue of necessity—that was, it reduced the effect of contact to a negligible item; this, in his opinion, was hardly the way to tackle turbine problems, and he would advise most exhaustable tests being made on cylinder distortion with a view to annuling the cause, and hence the effect would disappear, he hoped, nearly entirely. The finer the clearance the better the steam consumption, but there were other ways of improving the efficiency of turbines just as important as reduction of clearances, which were too often neglected; for instance, the frictional losses in the blading were a most important item, and could be considerably reduced by a proper attention to blading materials and construction. He ventured to submit a small chart, Fig. 18, which showed where the most important losses occurred, and these losses, if considered carefully, could be minimized considerably. Concerning the actual blading strip itself, this could be improved considerably, and it was hoped that figures would be given later which would explain more fully how a great saving in friction losses was obtained. There was another vitally important matter to be considered in the manufacture of turbine machinery if it was to reach the excellence of the present marine reciprocating engine, which was that men and methods would have to be revised and adapted to this special class of work. Turbine manufacture to be a success called for a much finer degree of accuracy and much more careful workmanship than was accustomed to be met with in the manufacture of big reciprocating machinery; while he was confident that designers and workmen would rise to the occasion, it was impossible to give too much attention to

this one of the most important factors making for the ultimate success of the marine steam turbine. There was an all-prevailing thirst for information on turbine design, but it must not be forgotten that the design is only the preliminary canter, as there were innumerable intricacies to be attended to in the manufacture and steaming which were, if anything, more important than the actual design. He sincerely hoped a good discussion would be the result of Mr. Speakman's valuable paper, and the great interest taken in steam turbines by the members would, he felt sure, result in an exchange of opinions invaluable to the records of the Institution.

Mr. John Ward (Vice-President) said it was a coincidence that most of the members who could speak with weight and authority on the subject were at the present moment (together with the members of the late Turbine Commission) guests of the builders and owners of the largest, and probably the most successful, turbine steamer yet built, viz: the *Carmania*. Her official trial had commenced on the Clyde that morning, and would continue until she arrived at Liverpool. It was an open secret that the trials of the steamer on the Clyde during the past week had fulfilled in every way the highest expectations of all connected with her. That was a matter of great gladness to every member of the Institution, but he also felt that it carried with it a tinge of sadness and regret, that the late Lord Inverclyde—the talented chief of the Cunard Company, who had such faith in the possibilities of turbine machinery, that he agreed to have it fitted in the *Carmania*, and so judge it on its merits in Atlantic running—had not lived to see the trials which were just concluded and had been so successful. The paper contributed by Mr. Speakman would, he was sure, prove of much value to the Institution. The author showed a thorough acquaintance with turbine construction, and many of the deductions he made from that knowledge and from the data regarding ships' performances, which he had collected, were of much interest. Speaking of turbine vessels built by Messrs. William Denny & Bros., much of the data given by Mr. Speakman could only be in the nature of an approximation, as none of it has been published by them, or others, as far as he knew. He did not, therefore, touch on the accuracy of the figures given, but there was one point which had not been taken up in the paper, and it was the question of power measurement in turbine vessels. While

it was not easy to indicate turbines in the ordinary sense of the word, it was possible to determine the shaft horsepower, namely, the power delivered to the propellers. The ability to do this, of course, gave them the power required to drive the vessel, and also the turbine efficiency, provided the steam consumed by the turbines was accurately measured. In addition to this, if one was provided with an experimental tank, extremely interesting results in screw problems could be obtained. For instance, were the effective horsepower of a vessel known, then by means of such an instrument as the torsion meter, now before them, and shown by the joint-pantentee (Mr. Charles Johnson), the net horsepower delivered to the propellers could be ascertained, which was usually called the shaft horsepower; and if the augmentation of resistance produced by the propellers were known—which could also be determined in the tank—a measure was at once had of the total screw efficiency. When his firm began to build turbine vessels, the need of such an instrument was at once apparent, and the question was gone into by their late President, Mr. Archibald Denny, and a member of the Leven Shipyard Electrical Staff (Mr. Charles Johnson). After a great expenditure of time and trouble, the many difficulties in the way were successfully overcome, and a practical and reliable instrument designed, and used with complete success on the turbine vessels built by his firm. To obtain the torsion, a suitable length of shaft was chosen for the measurement, and discs were fitted at each end. On each of these discs a permanent magnet was secured radially, with a sharp edge at the periphery of the disc, lying with the edge parallel to the shaft. Essentially there was a single-coil inductor set beneath one of the magents, and a series of coils beneath the other magnet. These were arranged circumferentially, and lay very closely together, so that the sharp edge of the magnet passed directly opposite each separate coil in turn. When the shaft was at rest, and without torsion, the magnet at one end was placed above its coil, and the magnet at the other end was brought exactly above the first of its series of coils. As soon as the shaft was revolved with the transmission of power, a torsion was put upon it. When the first magnet reached its single coil, it was apparent that the other magnet would no longer be above the first coils of its series, but above some other coil farther around the circumference. The recording box of the torsion meter was designed to ascertain which coil

was the one directly beneath the magnet at that time, and hence, knowing the spacing of the coils, the amount of torsion at the radius of their centers was ascertained. To obtain this information, the currents induced in the coils at each end were balanced, that was to say, when each magnet was above a coil the currents induced were adjusted by resistances, so as to exactly neutralize one another. A telephone receiver differentially wound was so connected that the separate currents from the inductors at either end of the shaft were made to act simultaneously upon the diaphragm of the receiver. There would only be one position where that neutralization took place perfectly. For all other positions a "tick" would be heard at every revolution of the shaft. By making contact with each coil of the series successively, it was easily discovered which was the particular coil where the neutralization took place, there being no "tick" for that position. This selection of coil was made by passing a switch over a series of studs, each stud connected to one of the series of coils, and each stud therefore representing a fixed amount of torsion, namely, the distance between adjacent coils of the series. A direct reading from the scale round the studs would give the amount of displacement of the magnets relatively to one another, and therefore the amount of torsion. The foregoing was a description of the torsion meter in its simplest form. In practice it was usual to supply six single coils at one end of the shaft, and a series of coils at the other, the distance between the single coils being rather less than the whole space occupied by the series of coils. This was done to give greater range of measurement to the instrument. In effect this arrangement multiplied the amplitude of measurement of the simple form about five-fold. The apparatus upon the table was fitted for three shafts, and, as would be seen, had two circles of studs for each shaft, one for the single coils and the other for the series of coils. There was provided a row of studs at the top of the box; these were connected to resistances, and were for balancing the currents induced by the two permanent magnets, which were not necessarily of exactly the same magnetic intensity. A feature of this induction method was that there was no actual touching of the fixed and revolving parts, and therefore no error introduced by the wearing away of surface. It could be left in position for any length of time, and was always ready for use. The recording box might be placed in any convenient position in

the ship where comparative quietness could be obtained to take the observations; the connecting cables being led to it from the inductors. He thought no apology was required for making this digression, as Mr. Speakman had pointed out how important it was that one should know the inter-relation between the turbines and the propellers. He placed a high value upon Mr. Speakman's paper, which would prove of service to members and a valuable acquistion to the Transactions.

Mr. Alex. Craig (Member) said he had read Mr. Speakman's paper with very great interest indeed, and he had noted several points upon which he would like to ask Mr. Speakman a few simple questions. With respect to the question of pressure per square inch of projected area of propellers, the author stated that pressure of from 12 to 14 pounds had been known, but he was under the impression that in some torpedo-boat destroyers the pressure reached as high as from 15 to 16 pounds. If this really was the case what percentage of slip of propeller might be expected under the circumstances? With regard to the adoption of turbines for tramp steamers, that would come, but in the meantime the most valuable information which Mr. Speakman, or any other expert, could put before them was in connection with channel steamers and destroyers. With regard to the ratio of projected area to disc area, the highest mentioned by Mr. Speakman was .58, but he knew that in some cases that figure had been exceeded. He had taken great interest in the turbine steamer *Victorian*, to which the author had referred in Table I, where it was stated that the propeller had a diameter of 8 feet 3 inches. To him the propeller looked larger than that, and he was certainly surprised to see that some material had evidently been removed from the tops of the blades. The original propellers were four bladed, and he understood that they were being replaced with others of three blades, about 8 feet 10 inches in diameter. He would like to ask Mr. Speakman what he thought about this change. Referring to Fig. 7, the author said that "Parsons turbines, however, have been built with *Vt* varying from .25 to .85 of *Vs*." He would like to know what this ratio was in channel steamers, and how this effected the economy of the turbine. In the formulae $(\text{Blade velocity},)^2 + \text{No. of rows} = \text{Constant}$, it was not quite clear what that constant really was, and the author in his reply, might, per-

haps, be able to state how it was used in turbine design. Coming to the blades of the turbine, Mr. Speakman seemed to think lightly of the principle on which the blades were fixed in the Parsons turbine. He (Mr. Craig) had seen a few Parsons turbines, and certainly to his mind the fixing of the blades seemed a weak point. The arrangement of blades exhibited on the table was a very excellent substitute for the method used by Mr. Parsons. The fixing of the blades was pretty much controlled by the weight of the hammer and the strength of the engineer's arm, but with this new method the variance of pitch of blades appeared to have been eliminated. From what he had seen of the Parsons turbine blades in position, they seemed not to be so regularly spaced as those in the Willans ring on the table. The action turbine was the turbine that he was practically wedded to, and he would like to ask Mr. Speakman if he thought that there was any possibility of the action turbine being used economically for driving steamships. It had certainly one advantage, in that superheated steam could be used. He did not know whether superheated steam had been used for Parsons marine turbines, but he looked to Mr. Speakman for this information. The stripping of blades would be a very serious matter if such occurred in midocean, and from what Mr. Speakman had said it was very apt to occur. But he gave no instance of such a thing having taken place, and many, like himself, would like to hear of the results of such stripping and the best way to obviate the same. The author referred to turbines for generating electricity and marine turbines, and said that the former were not designed for the same internal efficiency. What did the author mean by this internal efficiency? In Table III, he noticed that the widths of turbine blades varied from $\frac{3}{8}$ of an inch up to $1\frac{1}{4}$ inches, and he had been led to believe that for lengths of 21, 24 or 30 inches their widths should be greater. It would be seen from the diagram above, Table III, that there was a clearance at blade tips which extended for the full length of the space traversed by the steam without any opposition whatever. This clearance seemed to him to correspond with a leak in a main steam pipe, involving an irrecoverable loss. The retardation of the steam in its passage through the turbine by the wire lacing of the blades was also another loss which he thought might be obviated by a better mechanical device. In discussing matters of this nature, it was better to have reliable figures—he did not mean to say approximate

figures—but figures upon which an engineer could rely. He desired to thank Mr. Speakman for having submitted such a meritorious paper.

Mr. J. R. Jack (Member) remarked that Mr. Speakman's paper really concerned the engineer more than the naval architect, but as they had to work together it was of interest to them both. In taking up the question, Mr. Speakman hit the nail on the head at the very beginning, where he said that the turbine efficiency and the propeller efficiency must be considered separately and together. The propeller of a turbine-driven boat was nearly always too small, because if a big propeller was fitted a larger turbine would be necessary than the engineer could afford weight for. The same difficulty obtained, though on a smaller scale, in motor boats, and he thought the motor boat might be looked to for a solution of that problem. In such small craft, the propellers could be altered cheaply and experiments readily carried out. It appeared to him that the motor boat seldom gave a high efficiency, because the revolutions were kept down in order that the propeller might be large enough to have *some* efficiency; with the result that neither motor nor propeller reached their maximum efficiency, each having made a sacrifice to help the other, and so it was exactly on a par with the turbine vessel. Although he was afraid his engineering friends would scoff at the suggestion, he thought the time had arrived when the question of gearing should be considered. In the early days of screw propulsion, gearing was adopted, as the engines could not run fast enough for the propeller. Direct driving followed, and he felt that with the present style of motor or turbine it might be advisable to introduce gearing. He had experimented with gearing in a small way with a motor boat, and the results had surprised him. He tried a small petrol engine geared down two to one by a chain drive, with the result that the motor was able to run at its maximum revolutions. The propellers running at a reasonable speed could be of a reasonable diameter, and a ratio of effective horsepower to brake horsepower could be obtained almost as high as the best of the turbine boats that were mentioned in the paper. Of course, it was one thing to work gearing with a boat 25 feet long, and quite another matter with a ship 600 feet in length; but he thought there was a possibility of screw gearing being adopted somewhat

on the lines of that used in the Lanchester motor car. It would run for thousands of miles without showing appreciable wear, and such a gear might give the de Laval turbine a chance of being applied to marine propulsion. Another advantage of using a large propeller was in starting and reversing the vessel. With the small propeller, that was a weak point in a turbine ship, and a large propeller would get over the defect. Tank models helped largely in investigations, but they had their limitations. The principal limitation was atmospheric pressure. A model propeller placed a foot below the surface had an atmospheric pressure equivalent to nearly 30 feet of water on the top of it, and a propeller 30 feet in diameter had only its own diameter above it. That difference was so serious that it shifted the cavitation point out of scale altogether. To get over this difficulty, an experimental tank would have to be arranged with self-recording instruments the air being so rarified that during experiment it would bear the same relation to the model as the atmosphere did to the full-sized ship. There was another difficulty with the present type of turbine vessels. They had almost invariably three shafts. Now for river or cross-channel steamers that did not matter, but with large passenger and cargo vessels having such an arrangement the designer had a grievance. The bottom of the ship abaft the machinery space was monopolized by these shafts, and the result was that the center of gravity of the cargo spaces was very far forward, which, of course, could not be helped. The common center of gravity of the ship, when loaded, was much further forward than when light, and therefore inevitably there was a big difference of trim between these conditions. A vessel only devoted to cargo might be designed with a quarter deck or a long poop, or something of that kind, but cargo steamers driven by turbines were not yet in evidence—if ever they would be. Usually the upper part of a turbine vessel was monopolized by passenger space, and the objectionable feature could not be got over. If the center shaft could be got rid of, a tunnel as in an ordinary twin screw steamer might be arranged. There was another shipbuilding point referred to by the author regarding the defective ventilation of turbine rooms, and he knew that a great number of vessels suffered from that. It was a fault that was not altogether incurable, and it was largely due to the nervousness of the turbine maker with regard to stripping of blades and possible break down. A

turbine was long (unfortunately, it usually worked out rather longer than the ordinary engine), and it was liable to the disease of stripping, when it had to be taken out to be sent back to the maker; in consequence, the air casing had to be made large enough to let the turbine out. If the makers of turbines could guarantee that the turbines would not require to be unshipped for repairs, the shipbuilders could divide the air casing into two portions, with one portion over the after-end of the turbine room and the other over the fore end, and so have a complete circulation of air through the space. It was pretty much the same with regard to ships' galleys, which in some ships were very hot places. Some of them, indeed, were worse than any naval turbine room, in fact, it was almost impossible to live in them; but with this system of ventilation the galley nowadays was no worse than any other part of the ship. As to a comparison between ordinary screw steamers and turbine vessels, it was almost impossible to get data on which to work, because a comparison which was true today was not so tomorrow. The best published comparison was that respecting the Midland Railway Company's steamers, which furnished an excellent contrast between the *Antrim* and the *Londonderry*. That had been drawn up with every care to avoid undue favor to the turbine, and still the turbine showed a sensible advance on the reciprocating engines, taken at the time. But today that comparison was of little value. The *Antrim* was an exceedingly fine vessel, and if she had to be repeated today he doubted if either her designers or builders could improve upon her from the standpoint of speed, but with the *Londonderry* it was quite different. Coming from the 22 knots of the *Londonderry* to the 23 knots of the *Manxman*, there was an increase of boiler pressure of from 150 to 200 pounds. The turbines were designed to utilize all the steam produced by the boilers, whereas in the *Londonderry* they were designed to work best with the single-ended boilers shut down. In order to carry the extra weight, the *Manxman* had had her beam increased one foot. Today a *Londonderry* could be easily built to reach the speed of the *Manxman* without any of these increases, and in fact in the case of the *Invicta* it had been done on even smaller dimensions. He had looked through the paper in vain for one thing, and that was any reference to the gas turbine. The steam turbine had given the engineer a lighter motor—there was no doubt about that—its

introduction had got rid of a great deal of the vibration, if not all, but the weight and trouble of the boilers still remained. He was sorry to add to the number of questions which Mr. Speakman had to answer, but he would like to know if Mr. Speakman did not think that if the gas turbine did not come pretty soon it would have a hard fight with the reciprocating gas engine for marine propulsion.

Mr. W. H. Riddlesworth, M.Sc. (Associate Member), said it was with feelings of more than ordinary diffidence that he rose to address the meeting after the very eloquent opening by Mr. Ward. He would like to offer a few critical remarks on this interesting paper, but before doing so he would like to pay tribute to the skill with which Mr. Speakman had collected the tables and data on matters which were certainly not very easily got at. The first part of the paper was devoted to the relation of propeller and motor, and the intimate inter-relation of these had formed the subject of many essays and attempts at elucidation, whilst the attempts at overcoming the difficulties had produced almost as many wonderful devices as the search for the elusive perpetual motion machine.

The particular troubles incident to high rotational speed had often been made patent to those who had tried to obtain light-weight machinery or small dimensions of propeller. The steam turbine was essentially a high-speed motor, and it was only by a compromise between almost irreconcilables that any satisfactory propelling apparatus could be made. To that problem the first part of the paper was devoted, and it was assumed as a working basis that the desirable thing was to make the revolutions as great as was consistent with a reasonable propeller efficiency. That was a point which required some proof, as it was quite conceivable that some lower speed at which the propeller efficiency might be greater would pay, in spite of the probably increased dimensions and weight of the turbines. The author's first step in the solution of this part of the programme was to obtain the minimum diameter of the propeller, and this he did without reference either to pitch or revolutions, both of which must obviously have no inconsiderable influence on the dimensions of the propeller. He would suggest that a somewhat similar formula to that on page 48 might safely be used to give the minimum projected blade area, but that in this present form the author's

formula was incomplete and apt to be misleading. It was probable for ships which had to make fair weather passages and for which a high trial trip (in smooth water) record was the desideratum, that the pressure should be pushed to the point at which cavitation was imminent, but for ships that had to keep time in all weathers the pressure allowed should be kept down something like in the ratio of the smooth-water resistance to the augmented resistance due to bad weather, etc. On page 51 the author said: "Having obtained the diameter of the propeller and the revolutions possible, the design of the turbine can then be undertaken, but for this no formulae exist at present, such as are met with in reciprocating-engine practice." He had tried to show that the diameter obtained by the formulæ on page 48 ought rather to be regarded as an indication of the minimum projected area, whence, of course, assuming a surface ratio, diameter could be obtained, but he nowhere saw any method of determining revolutions unless he was to infer it from the allowable speed of blade tips.

The relation between axial thrust and circumferential speed of blade was by no means a simple one, as might be seen from the author's Fig. 1, which might be accepted for the present purpose. Hence any calculation of revolutions from these two elements alone must be of somewhat doubtful value. Accepting the thesis that the revolutions must be kept as great as practicable, he would suggest that the propeller, should be designed on the ordinary lines, if possible, by a method involving all the elements of the propeller; that was revolutions, diameter, pitch ratio, and surface ratio, etc. That might have to be done for a series of revolutions, and one ought to be able then to see what sacrifice of efficiency would be necessary in any particular case in order to keep the thrust pressure on the blades below the predetermined maximum. The maximum revolutions could also be determined, and the problem solved generally. Even then, the solution or solutions of the propeller problem would not be as definite as might be wished, as it was well known that many cases have arisen in which the best information had been read as a basis for determining propeller dimensions, and yet vast improvements have been made by changes of propeller which had apparently not been foreseen in the original solution. The case of the cruisers of the *Drake* class would at once occur to many. Assuming the dimensions of the propeller fixed, then the next consideration was of the turbine problem.

He did not propose to deal with that at any great length, but he must call the author's attention to the looseness with which he had explained his method of finding the velocity of flow of steam. That was, he thought, best explained as that velocity which implied that the added kinetic energy of the fluid was equal to the work done in expanding, in whatever way it did expand—adiabatic or otherwise—from the initial pressure to the final pressure in the passage under consideration.

In Fig. 18 A, if a pound of steam at $p_1 v_1$ passing through the turbine expanded to $p_2 v_2$, then at any particular stage in the turbine, when if the pressure was p v for a small drop in pressure, such as from row to row, the work $= v\,\delta p$, and the velocity was such that $\frac{V_s^2}{2g} = -v\,\delta p$, the initial velocity being neglected, or with an initial velocity $V_i \frac{V_s^2 - V_i^2}{2g} = -v\,\delta p$.

In a paper read before the Institution last session, Mr. Melencovich gave a very elegant method of finding the successive pressures, and hence the drop in pressure from row to row when the work was equally divided amongst any

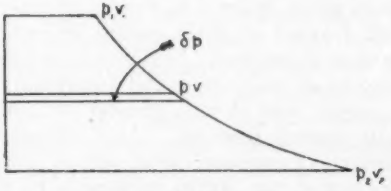


Fig. 18 A.

number of rows of blades. It might be noted that the author's use of "8" as the value of $\sqrt{2g}$ in his formula was capable of a refinement which was surely well worth making. Even a common 10-inch slide rule showed a very considerable difference between the accepted value of $\sqrt{2g}$ and 8. In connection with the relation of the blade velocity and the steam velocity, it was worthy of note that the two cases quoted at the middle of page 53 were only correct when the angle between direction of motion and the impinging or issuing jet and the blade was zero; for all other cases, a relation depending on this angle held good. Of course, in a

combination impulse-reaction turbine the relation of these velocities was empirical to a certain extent. In conclusion, he would point out the futility of elaborate theoretical investigations of the necessary blade heights, &c., when it was impracticable to approximate to these dimensions, and the blades had to be made of constant size for very considerable portions of the length of the turbine, although by suitably grading the packing pieces much might be done to make the openings of approximate sizes, and until a method of building turbines was evolved which allowed each row of blades to be made of the proper size, all turbine dimensions must of necessity depend on a number of empirical rules which would naturally be guarded with the greatest jealousy by the fortunate possessors of sufficient experience to enable them to devise such rules. One line of progress, it seemed to him, lay in the direction of abolishing steps in the size of the various rows of blades, and another was in the adoption of a more mechanical construction of the blades and passages, one example of which was shown by a very enthusiastic advocate at the last meeting.

Mr. John Reikie (Member) considered Mr. Speakman's paper one of very great interest to every engineer. Although the author had framed it with special reference to marine work, the question of economy in steam consumption was of vital importance to all engineers, and any remarks bearing on that question might lead to still greater economy than that now attained in general practice. As a locomotive engineer —in which branch of engineering it was at present impracticable to make use of the steam turbine — he did not agree with Mr. Izod that this type of engine was the only one for future marine work. For marine practice, although the turbine might at present have a great advantage over that of the slow-speed triple-expansion reciprocating engine with respect to high rotative shaft speed and steady running, still the introduction of a high-speed reciprocating engine in combination with a high steam pressure of 500 pounds and upwards, and designed so as to make use of steam superheated to a high degree, would not only place it on an equal footing with the turbine as regards high rotative speed and eliminating vibration, but it would enable it to reach a very much lower consumption of steam than that possible with the turbine as now designed. Regarding economy in steam consumption, his

opinion was that if the maximum of efficiency was to be attained with every engine designed, it became necessary to have a standard in the quality of steam generated in the boiler, equally as much as for material used in making the engine. So far as he could judge, the highest steam pressures carried in boilers were accepted by many engineers as being the best in quality. That appeared to him to be a mistaken idea; the measure of quality in steam should be temperature, not pressure, so that every engine should compete on equal terms in economy in steam consumption, independent of the pressure carried in the boiler. On page 36 the author, when referring to the cylinder of the turbine, mentioned a temperature of 400 degrees as a maximum, although that temperature did not admit of the best quality of steam being used in the cylinders, still it would be found by a reference to Fig. 12, that the consumption of steam was as low as 13½ pounds per I.H.P. He would like to ask the author if the steam used in the cylinders of the engines of the other vessels shown in the same table was of the same temperature, for if not it might go a long way to explain the difference in steam consumption. As to the introduction of a standard quality of steam, experiments only would bring to light what the temperature should be to form that standard. So far he had found no difficulty from using a temperature as high as 900 degrees in cylinders; this, however, might be as much above the average required to bring about the best results as 400 degrees was too low. Experiments on the Continent with a 300-H.P. engine had proved that with a temperature of about 540 degrees Fahrenheit, the consumption of steam per I.H.P. was brought down to 9.9 pounds. What could be accomplished with one engine should be within the reach of all engineers; a universal reduction of steam consumption was surely a goal all interested in the search for economy should endeavor to reach. If the steam turbine could not compete with the reciprocating engine under such exacting conditions, he had no doubt it could be designed to do so. He was not quite sure if Mr. Izod had not already in view the early introduction of very highly superheated steam, for he remarked that the turbine of ten years hence might differ materially from that of the present day.

Mr. James Hamilton (Member) remarked that some observations made by Mr. Jack had suggested to him that he might

say a word or two which might be useful in connection with cavitation and the figures for the pressure per unit of projected area of propeller. He had for a long time thought that blade interference had more to do with inefficiency of propellers than cavitation, and he had taken the trouble to plot down the path traced by each blade of a propeller, and it was very surprising to see, when allowances for the advance of the screw, the slip, and the wake were made, what a very small layer of water each blade had to work upon with the fast vessels, such as were under discussion, and it was worse when the ship did not travel fast like the screw. That had been confirmed in a small yacht fitted with a motor which he owned, and he was surprised to find that in making a distance of about 13 nautical miles, after one of the blades of the propeller had been thrown off, he went about as fast with the one blade as with the two blades. It was not very good for the shaft, but he did not think it suffered. The point was that there was a great deal less blade interference, and there was very little difference between the speed with the two blades and the speed with the one blade. Of course the engine was running about 1,000 revolutions in the one case, and about 750, or the normal speed, in the other case, the propeller going at half the speed of the engine in each case, connection being by a chain belt and pulleys. Those investigating propeller problems should consider the question of blade interference. It was perhaps easy to calculate the pressure on the disc, but in what way could an analysis be made to show that after reaching 11 or 12 pounds per square inch the efficiency fell off? Although the adoption of such a pressure might be a convenient way of ensuring that the propellers were not made too small, the falling off in efficiency might be equally a question of blade interference and not a matter of cavitation at all.

Mr. James Anderson (Associate Member) said that Mr. Speakman had stated that the tensional pressure of propeller blades was approximately from 10 to 12 pounds per square inch at a depth of 12 inches below the surface, and he desired to know if the indicated horsepower used in the thrust formula, got from model experiments, was calculated from the steam consumption or from a torsion meter. He considered the formula at the foot of page 53 somewhat indefinite, and he could not make out what "8" was.

Mr. John Alexander (Member) asked the author if he would tell the Institution what the present method of grouping the turbines was in a vessel running at cruising speed. In electrical work the power of the steam turbine was generally governed by alternating the period of the blasts of steam admitted to the turbine, and he would like to know if this method was used in vessels with turbines running at reduced speeds.

Mr. R. T. Napier (Member) observed that the use of triple propellers came up in connection with the application of the steam turbine to marine purposes. Those who had experience with twin screw steamers knew that the slip was much greater than with single screws, and he asked, if it was not a business secret, whether it was customary to make the center propeller, where three were used, of less pitch than the wing propellers. If this was not done at present, he was prepared to learn some day that power was being lost. Another matter was the time required for the construction of large steam turbines. The number of blades on a single motor amounted to hundreds of thousands, and the number that could be fixed in a working day was limited. He asked if it was not the case that for engines of large power those of the turbine type required longer time to construct than did multi-cylinder engines.

Mr. Robert Royds, B.Sc. (Member), observerd that Mr. Speakman had said that the greater the ratio of *Vt* to *Vs*, the greater would be the required number of rows of blades. It appeared to him that the number of rows required depended on the amount of useful work that could be given to the turbine by the steam per row of blades. He would like to know what particular law Mr. Speakman adopted when he calculated the volume for actual expansion, whether it was with 70 per cent. efficiency or 80 per cent., or some intermediate value.

CORRESPONDENCE.

Mr. R. M. Neilson (Manchester) felt that Mr. Speakman's paper was full of useful information, much of which, he believed, had not been published before. Mr. Speakman had referred to the leakage over the tips of the blades in Parsons' turbine. The effect which that leakage had on the efficiency of a turbine was a question which had been generally pass-

ed over, without comment, by writers of papers on steam turbines. That was probably due to the fact that it was difficult to tell how much loss was due to this cause. Mr. Speakman suggested that that leakage was perhaps not so detrimental on account of actual leakage loss as in its heating effect, because the heating of the steam increased its volume, and thus upset calculations regarding dimensions. It was, of course, impossible in designing a turbine to calculate exactly what the heating effect due to leakage over the tips of the blades would be; but it was quite possible, he believed, to calculate the heating effect with sufficient exactness to enable the turbine dimensions to be determined without great error due to that cause. The changes of volume of the steam caused by friction and by the transference of heat to the metal parts were, in his opinion, much more difficult to ascertain beforehand than the change of volume due to the heating effect of leakage past the tips of blades. While the difficulty due to this change of volume, produced by leakage past the tips of blades, was, in his opinion, not serious, he believed that the direct loss due to this leakage might be, and in some cases was, very serious. A certain amount of heat energy of the steam was at every stage converted into kinetic energy. With the kinetic energy so acquired, some of the steam—sometimes a large percentage—leaked past the tips of the blades, and a great part of its kinetic energy was converted back into heat. It might be thought at first sight that this energy was not lost—it was only converted into another form. As a matter of fact, however, *much* of it is lost as far as ultimately getting useful work out of it was concerned. This would be obvious on a little consideration. He did not need to give the proof here, but the loss was on the same lines as that which would occur if the heat produced by friction in the turbine bearings were used to generate steam to drive the turbine. The leakage past the tips of the blades was, in his opinion, an important loss in turbines of the Parsons type. The percentage of leakage was usually greater at the high-pressure end of the turbine, but the thermodynamic loss for a given weight of leaking steam was here least. Fig. 9, showing the tip clearances, was very interesting, but its value would, he thought, be much more added to if the heights of the blades were also given. He presumed that the overall diameter was measured from tip to tip of the rotating blades, and was not the outside diameter of the fixed

casing. A little more explanation of Fig. 10 would also be useful. Mr. Speakman gave a formula, which depended on the effective thrust, for finding the diameter of a propeller. In order to use that formula as the author intended, it was necessary to know what assumptions he had made in obtaining the effective thrust. On page 57, it was stated that "the higher speed necessary on the L.P. was got by running at considerably higher revolutions on the H.P. shaft." Why "necessary"? In referring to the material of which turbine blades were made, Mr. Speakman said the alloy contained 16 per cent. of copper and 3 per cent. of tin. It would be interesting to know what the remaining 81 per cent. was.

Mr. Henry L. S. Nicol (Southampton) stated that having had a considerable experience in turbine work, both in designing and running turbines from 200 kw. to 5,500 kw., and being at present with Messrs. Thornycroft, directing their turbine work for torpedo-boat destroyers, he felt much indebted to the author for his very able and valuable paper. One had only to study the subject a little to find the many difficulties which had to be overcome before the final design could be attempted. The author had, as one might say, laid the foundation stone. Of the many papers which had been read before the various Institutions, perhaps never had the subject been so openly treated, and he hoped the formulæ given in this paper might be fully considered and bear a future weight in turbine design. From a practical point of view he would like to bring forward three important points, namely: (1) distortion through cylinder design, (2) balancing, and (3) end thrust. With regard to distortion, he had met with it in several turbines, and in one special case, an 1,800 kw. machine on test (after being balanced, and all other preliminaries attended to) commenced to tip so badly that the test had to be abandoned. That was due to the distortion of the cylinder, and it had to be overcome by "relining up" the spindle and increasing the blade-tip clearance between the cylinder blades. That was done by fixing and driving a narrow emery wheel, to an arrangement on the boring bar, keeping the bar out of its true center, thus increasing the spindle-blade tip clearance where actually required, and treating the cylinder-blade tips in the same way. Although he approved of superheated steam, naturally such high temperatures caused more distortion. This distortion,

he was afraid, would never be totally overcome, as it took place in various directions. The most effectual way, in his experience, had been to increase the number of radial ribs or webs.

As to balancing, that was also a very important point, and they had recourse to the knife edge, which should have plenty of bearing surface to ensure easy rolling and to prevent marking heavy spindles. Then, finally running the spindle in its bearings, either by a motor, apart from the turbine, or by steam in its cylinder. The provisions for fixing balance weights, the number and, easy access to same, were points in design of great advantage. The end thrust, the author said, could be calculated, and great care was necessary in the calculation. The figures could only help to decrease excessive end thrust, the final end thrust being acquired by careful measuring and regauging at the blades.

Mr. John Ward (Vice President) stated that it was incorrect to say that the torsion meter was not a dynamometer. Perhaps Mr. Napier had in his mind the kind of dynamometer which absorbed the power to be measured. It might be pointed out that the torsion meter was greatly superior to that kind of dynamometer, in that it measured the power transmitted by a shaft while it was doing useful work. Every kind of dynamometer must of course be calibrated. In the case of the torsion meter, the shaft formed an essential part of the whole dynamometric apparatus. It was the spring which had to be calibrated, and this could easily be done before it was placed on the vessel, or, alternatively, it would be found fairly accurate to use the following formula for ordinary solid-steel shafts:

$$WR = \frac{140 \times (d)^4 \times \theta}{L};$$

Where WR = foot pounds turning moment.
d = diameter in inches of shaft.
θ = angular deflection in degrees. } Between the discs of the meter.
L = length in feet of shaft. }

Mr. William Gray (Member) stated that the figures which Mr. Speakman quoted from his paper, read at the Institution of Naval Architects in July last, were based on observations made at the speed trials of the respective vessels, and they

were not approximate. He (Mr. Gray) did not say they were absolute figures, they were given in the paper as comparative, with the word "comparative" italicised. The *Manxman* had one foot more beam to give her increased stability, as she was built for the Isle of Man passenger traffic, and the whole of the shade deck was available for passengers as compared with a portion at the forward end in the other vessels. The increase in beam had no connection with the larger turbines.

Mr. John H. Macalpine (Member) stated that he had read Mr. Speakman's paper repeatedly with interest and profit, and he thought there were quite a number of points which the members of the Institution would like the author to amplify. Some of these he would touch upon in the following remarks. The co-ordinate subjects of propulsive efficiency and propeller thrust were frequently referred to. But while on page 51 effective thrust was referred to as a somewhat subtle subject, at quite a number of places the true value of the propulsive efficiency or thrust was supposed to be known, or to be calculable with considerable accuracy for the various classes of work, when determining the proper dimensions of the propeller and the turbine. In referring to Table I, the value of C was stated to be probably within 2 per cent. of the truth. As C was proportional, in any case, to the square root of the effective thrust, the value of this thrust would then be true within 4 per cent. It was hardly to be supposed that anything like such a close approximation could be arrived at except by a most careful experiment on the actual ship or on a model. In the latter case, when passing from the model results to those of the ship, with a propeller under a pressure of 8 to 10 pounds per square inch of blade area, the margin of possible error was considerable; for while the full-size propeller was near the limit at which it would work without cavitation, the intensity of pressure on the small model propeller, at the corresponding speed, was quite light, and it was not to be expected that the action of the water in the two cases would be entirely similar. If the full-size propeller was producing cavitation, the action would be very dissimilar. This was a question that would repay careful experimental research. To predict, in a proposed design of ship and propeller, the value of the effective thrust within 4 per cent. seemed to him practically impossible, and in the design, both of the propeller and turbine,

much larger errors would have to be allowed for. But the question of propulsive efficiency was one to which Mr. Speakman had evidently given extended study, and he felt sure that he could add much of interest to what was popularly known on the subject, and give interesting data on which his conclusions were based. In Table I the approximate value of C was given as 30.8. From the equation—

$$\text{Diameter of propeller in feet} = \frac{\sqrt{T}}{C};$$

$C^2 = \frac{\pi}{4} \times$ effective thrust per square foot of disc area.

From this, for the *Amethyst*, effective thrust per square inch of disc area—

$$= \frac{30.8^2 \times 4}{\pi \times 144} = 8.4 \text{ pounds}, \quad . \quad . \quad . \quad . \quad (A)$$

In "Engineering" of November 18th, 1904, would be found the following data for the *Amethyst:*

14,000 I.H.P. (This must be estimated.)

Propellers.	Diameter.		Pitch.	Revolutions per minute.
	feet.	*inches.*	*feet.*	
Center,	6	8	6.56	449.4
Starboard,	6	8	5.75	484
Port,	6	8	5.75	499

From which—

Travel of propellers per minute.

Center,	6.56×449.4	$= 2,948$
Starboard,	5.75×484	$= 2,783$
Port,	5.75×490	$= 2,869$
	3)	8,600
	Mean,	2,867

From this the *indicated* thrust approximately—

$$= \frac{14,000 \times 33,000}{2,867} = 161,000 \text{ pounds.}$$

The total disc area $= 34.9 \times 3 \times 144$ square inches. Therefore, the pressure per square inch for 100 per cent. propulsive efficiency—

$$= \frac{161{,}000}{34{\cdot}9 \times 3 \times 144} = 10{\cdot}7 \text{ pounds.}$$

Equation A gave the effective thrust per square inch from Mr. Speakman's figures, from which the propulsive coefficient was—

$$\frac{8{\cdot}4 \times 100}{10{\cdot}7} = 78{\cdot}5.$$

There must be some serious discrepancy between the data used in the paper and that given by "Engineering," as it seemed hardly likely, with propellers working under such high pressure, that the propulsive coefficient was much, if any, above 50. The sentence, "It seems likely, to some at present undeterminable extent, but within narrow limits for each class of vessel, that the propeller efficiency is proportional to the coefficient *C*, and this seems to be borne out by the trials of the *Manxman* and the *Londonderry*," he found very obscure, and would like Mr. Speakman to explain it more fully. As *C* was proportional to the intensity of disc pressure, it seemed to him that its relation to the efficiency of the propeller must be quite complex, involving a number of other variables, and following a law very different from the simple one stated. No doubt Mr. Speakman's conclusion that where there was danger of cavitation there was no advantage of departing from a true screw, was founded on good data; but it seemed obvious that with a true screw the pressure at the tip must be a good deal more intense than at the root of the blade. This might be a positive advantage for low-pressure screws, but surely must be a disadvantage when the pressure was near the allowable limit. This seemed to be borne out by the experience of the writer with some torpedo-boat propellers. In one set, the variation of pitch was such as to make one expect great irregularity of pressure and consequent bad performance at high speeds. These were replaced by propellers which had a diminished pitch towards the tip. In the first, the breakdown of efficiency occurred long before the maximum speed was attained, and in the second, there was no indication of breakdown, though the contract speed was exceeded by fully 1½ knots. The fining of pitch towards the tip, giving diminished slip and more edgewise movement through the water, must increase the peripheral speed which could be used without producing cavitation.

With regard to the turbine, also, he was sure Mr. Speakman could readily add much of very great interest. He suggested that Mr. Speakman should give an appendix containing the full calculations for a particular case, giving both the rough calculations on which one would make the preliminary determination of blading and the more exact calculations of fall of pressure, efficiency, etc., with notes at the various steps showing what was deduced from theory and what from experience and judgment. Perhaps this was asking too much, but it would compress a large amount of information into a concise and easily used form. All who had had to do with turbine calculations must know that the theory was very imperfect, and experience counted for a great deal. So far, those who had had most experience, especially with marine turbines, were exceedingly niggardly in what they had put on record, and Mr. Speakman would render a distinct service by boldly breaking this reserve. There was much which one would like to see discussed further. For instance, it was stated that "the best ratio of blade speed to steam speed was a matter of opinion." Was it not the case that $V_t / V_s = .6$, as used frequently in electrical work, had been found to be the best, but that this ratio must be reduced in marine work partly, as Mr. Speakman stated, to keep down the number of rows of blades; but still more, at some sacrifice of efficiency, to accommodate the fast moving turbine to the slow moving propeller? A few lines would have made clear the origin of the important formula for equal powers and efficiencies—

$$(\text{Blade velocity})^2 \times \text{No. of rows} = \text{Constant}.$$

For if V_t / V_s = constant, the flow (neglecting secondary losses) would remain similar for the different values of V_s; and the loss of head in passing each row would be proportional to V_s^2 (or V_t^2). The similarity of flow also showed that the steam, on passing one set of blades, was deprived of energy in proportion to V_s^2; hence, for equal powers, the larger V_s^2 (or V_t^2) the smaller, proportionally, was the number of rows required. Also, the total loss of head would remain constant. It seemed probable that the eddying and other losses would not greatly alter this constancy of efficiency. The tabulation of the value of the constant of this formula in a few typical cases would be very valuable. He thought Mr. Speakman had not stated the real difficulty which prevented the turbine being adopted in slow ships. The propeller must be large enough to drive the ship against a head wind and

sea, or to keep it off a lee shore. This, together with any allowable pitch ratio, would bring the revolutions so low that the turbine could not be thought of. Fig. 10 showed the expansion line of the steam lying between that for saturation and the adiabatic line, as of course it should since, due to imperfect efficiency arising from frictional and leakage losses, etc., less energy was extracted from the steam between the two definite temperatures than would be were the expansion exactly adiabatic. Thus the actual expansion curve, such as that given in Fig. 10, would not be quite the same for all turbines. It would be of interest if the author set down exactly the assumptions made in this particular case. It was obviously a point of considerable importance in the determination of blade openings and the intelligent design of the turbine.

Prof. A. Jamison (Member) stated that Mr. John Ward had showed and explained the construction and action of the very interesting and ingenious Denny-Johnson torsion meter which his firm had used; but he did not give the full particulars concerning the percentage accuracy of the results derived from it, nor refer to the reliance that could be placed upon these particulars. Further, members would feel greatly indebted to Mr. Ward if he would give them details of the necessary preliminary tests of shafts in the works, for angle of twist produced by different torques, upon different sizes and qualities of shafts. It would also be interesting to know how closely the results arrived at by use of the formula Mr. Ward gave, agreed with the results obtained from the torsion meter, since he (Prof. Jamison) had failed to prove that the constant 140, used in Mr. Ward's formula, agreed with the constants given by various authors, as shown by the accompanying remarks. The formula given by Mr. Ward for the turning moment of a screw-propeller shaft was as follows:

$$WR = \frac{140 \times (d)^4 \times \theta}{L};$$

Where WR = Foot-pounds turning moment.
d = Diameter in inches of shaft.
θ = Angular deflection in degrees. } Between the discs of the meter.
L = Length in feet of shaft. }

It, however, did not agree with the formula deduced from Seaton, Bauer, Ewing, and Jamison's rules for the torsional

moment of resistance of circular solid shafts, which all closely agreed with each other, viz: the turning or twisting moment.

$$T.M. = \frac{C\pi d^4\theta}{32L} = \frac{Cd^4\theta}{10.2L} = \frac{515d^4\theta}{L} \text{ inch-tons.}$$

Where C = modulus of rigidity, was taken as 5,250 tons per square inch.
d = diameter of shaft in inches.
θ = the total angle of torsion or twist in circular measure.
L = length of shaft in inches between the two fixed points.

Whereas, in Mr. Ward's formula—

$$T.M. = \frac{140 \times d^4 \times \theta}{L} \text{ foot-pounds.}$$

But θ in degrees was equal to $\theta \frac{180}{\pi}$ in circular measure.

Hence,

$$T.M. = \frac{140 \times d^4 \times \theta \times 180 \times 7 \times 12}{L \times 12 \times 22 \times 2{,}240} = \frac{3.6 \times d^4 \times \theta}{L} \text{ inch-tons,}$$

when d = diameter of shaft in inches.
θ = the total angle of torsion in circular measure.
L = length of shaft in inches between the discs.

Would Mr. Ward kindly explain wherein this great, apparent difference occurred between the numerical constants 3.6 from his formula, and 515 as derived from the other authorities mentioned above? It might be as well to state that Bauer and Ewing used a value for the modulus of rigidity, $C = 5{,}250$ tons per square inch. Also, they used a shearing stress in the case of steady motion, $f_s = 13{,}500$ pounds per square inch with steel; whilst, with wrought iron, $f_s = 9{,}000$ pounds per square inch, which might be taken as safe values for the stresses in shafting.

Mr. Robert Johnson (Associate Member) considered that the results so far achieved reflected the greatest credit on those responsible for the introduction of the steam turbine for marine propulsion, but it was a matter for regret that more had not been made public both with regard to the performance of the machinery and to the method of design. The advantages claimed for the turbine might be considered under

four heads: 1, absence of vibration; 2, reduction in space occupied; 3, reduction in weight, and 4, increased economy. As to the first, it was a temptation to think that this point had been overestimated. So much had been said about it, and the subject had been forced into so much prominence that, one was led to believe that the bogey of vibration had been set up for the express purpose of being knocked down again. His own practical experience with turbines certainly did not confirm the assertion that less space was occupied, *i.e.*, less length of ship (the important matter in the case), since in all cases with which he had been brought into direct contact, the fore-and-aft length of engine room would easily have sufficed for reciprocating engines of the same power.

In certain classes of vessels where the speed of rotation of the machinery was high, everything was sacrificed to obtain lightness. Saving of weight was certainly an important matter, but from experience so far gathered with turbine machinery of the largest description, any material reduction in weight would not be consistent with durability and reasonable propeller efficiency, and the natural tendency at present was to make the turbine more substantial. It was upon this question of economy that the marine turbine must stand or fall. Unless the turbine could show itself *cæteris paribus* superior in economy when employed for propulsion, it could never supersede the piston engine for the ordinary purposes of commerce. And it was curious that those who knew the facts and who appeared to be most directly interested in the matter maintained a discreet silence on this point. The case of H.M.S. *Amethyst* was perhaps a striking exception, but it must be remembered that extraordinary methods were adopted to secure efficiency, and the ship carried no fewer than seven turbines, three only of which were required for full speed. This involved a degree of complication and an amount of expense which could not be seriously entertained as a permanent condition. As to the contention raised with regard to reversing, viz: that a turbine-propelled ship could be brought to rest in the same distance as was required for a ship with ordinary engines, it was exceedingly difficult to believe that the astern turbines would take effect on the ship as quickly as piston engines, in view of the fact, inseparable from all turbines, that of necessity the starting torque was small. Mention was made in the paper of cases in which the blades of turbines had been stripped. This appeared to be a

not altogether infrequent occurrence, and would be a very serious matter especially if it happened in mid-Atlantic. Blade stripping might arise through the very small radial clearance permissible; if so, then a hot bearing would almost of necessity be followed by the loss of many blades. It was a matter for regret that Mr. Speakman had not been able to enter into the question of design more fully, but nevertheless the paper formed a substantial addition to the literature of the subject. It would be of great interest if the author could supplement the information already given by stating what the angles of entrance and exit respectively of the standard blade section really were. Mr. Speakman said "The actual volume of the steam—not the volume per pound, as found in tables, or the volume due to adiabatic expansion, but the exact volume per pound at any point along the turbine, etc." He was quite at a loss to know what this meant, as the only volume the steam could have per pound was the volume as per tables, the suitable table of course being used. Was it possible that Mr. Speakman meant the volume of steam, per pound of *mixture*, at any point, etc.?

Comment by Ernest N. Janson, M. E.

U. S. Navy Department, Member American Society Naval Engineers.

The following formulæ $V_s = 223\sqrt{B.T.U.}$ as given in Mr. Speakman's paper reprinted in this issue of the Journal needs, in the opinion of the writer, some modification if intended for calculation of steam speeds through the vanes of a Parsons turbine. V_s in the above formulæ represents the steam speed at the exit of both guide and rotating vanes, and, although this varies in the different stages throughout the turbine, we will, for the sake of more ready illustration, consider it constant. This steam speed is dependable not only upon the "heat drop" in a particular row of vanes, but also upon the "heat drop" which has taken place in the next previous row, and due to which the steam entering the next following row, as a result of expansion, is given a certain velocity W_s, which varies with the B.T.U. "heat drop" as well as the peripheral vane spade. The entering steam speed W_s, commonly called the relative velocity, in any row of vanes constitutes the one component in the speed triangle in which V_t, the peripheral vane speed, makes the other, and V_s, the absolute exit velocity, forms the resultant.

Date.	Vessel.	Service.	Owner.
1894 Rebuilt 1896	Turbinia	Experimental	C. A. Parsons
1900	King Edward	Pleasure Steamer	Turbine Steamers Limi
1901	Queen Alexandra	,,	,,
1898	Viper	T.B.D.	R. N.
,,	Cobra	,,	,,
1903	Velox	,,	,,
1904	Eden	,,	,,
1905	Coastal Destroyers	,,	,,
,,	Ocean-going do.	,,	,,
,,	Experimental do.	,,	,,
1903	Tarantula	S.Y.	W. K. Vanderbilt
,,	Lorena	,,	A. L. Barbour
,,	Emerald	,,	Sir C. Furness
1905	Albion	,,	Sir G. Newnes
1905	Narcissus	,,	A. E. Mundy
1905	Royal Yacht	,,	H.M. King Edward
1905	Mahroussah	,,	The Khedive of Egyp
1903	The Queen	Channel Steamer	S.E. & Chatham Railway
,,	Brighton	,, ,,	L.B. & South-Coast Rly.
1904	Princess Maud	,, ,,	Stranraer and Larne Ser
,,	Londonderry	,, ,,	Midland Railway Co.
,,	Manxman	,, ,,	,, ,,
1905	Viking	,, ,,	Isle of Man S.S. Compa
,,	Onward	,, ,,	S.E. & Chatham Railwa
,,	Dieppe	,, ,,	L.B. and South-Coast Rly
,,	...	,, ,,	G. & J. Burns
,,	...	,, ,,	Great Western Railway
,,	Princess Elizabeth	,, ,,	Belgian Government
,,	...	,, ,,	Hamburg Heligoland S.S
1904	Lhassa	Persian Gulf to India. Intermediate	British India S.S. Comp
,,	Loongana	Inter-Colonial Service, Tasmania—Melbourne	Union S.S. Co. of New Ze
,,	Turbinia II.	Pleasure Steamer. Lake Ontario	Turbine S.S. Compan
1905	Maheno	Inter-Colonial	Union S.S. Co. of New Ze
,,	Bingera	Australian Passenger	,,
,,	Victorian	Atlantic Intermediate Service	Allan S.S. Company
,,	Carmania	Atlantic Mail	Cunard Company
1904	New Cunarders	,,	,, ,,
,,	Amethyst	3rd Class Cruiser	R.N.
1905	Lubeck	,,	German Navy
,,	Salem	Scout Cruiser	U.S.N.
,,	Chester	,,	,,
,,	Dreadnought	Battleship	R.N.
,,	Orion Class	Armoured Cruisers	,,
	No. 243	Experimental Torpedo Boat	French Navy
	Libellule	,, ,,	,, ,,
1903	Caroline	,, ,,	,, ,,
1904	No. 293	Torpedo Boat	,, ,,
,,	No. 294	,,	,, ,,
1905	S. 125	T. B. D.	German Navy
	Revolution	Experimental S.Y.	Curtis Marine Turbine Con

TURBINE STEAMERS—GENERAL DIMENSIONS AND DATA.

Owner.	Builder.	Length	Beam	Depth	Draught	Speed	Equivalent I.H.P.	No. of Shafts.	Screws per Shaft.	R.P.
		Ft. Ins.	Ft. Ins.	Ft. Ins.	Ft. Ins.	Knots.				
A. Parsons	C. A. Parsons	100	9	—	3	32	2000	3	3	23
Steamers Limited	... Denny Bros. ...	250	30	10 - 6	6	20·48	3500	3	{ 1 Centre 2 Wing	{ 503 / 730
,,	... ,, ...	270	32	11 - 6	6 - 6	21·43	4400	3	1	{ 730 / 1000
R. N. ...	Hawthorn, Leslie & Co.	210	21	12 - 9	6 - 9	36·58	13000	4	2	11
,, ...	Armstrong, Whitworth & Co.	223	20 - 6	13 - 6	7 - 3	30·2	10000	4	3	10
,, ...	Hawthorn, Leslie & Co.	210	21	12 - 9	7 - 3	27·1	7000	4	1	8
,, ...	,,	220	23 - 6	14 - 3	8 - 3	26·2	7500	3	2	9
,, ...	Thornycroft, Yarrow, & White	175				26	3600	3	1	12
,, ...	{ Laird, Thornycroft, Armstrong, White, Hawthorn, and Leslie & Co. }	250				33	15000	3	1	7
,, ...		320				36	28000	4	1	6
K. Vanderbilt	... Yarrow ...	152 - 6	15 - 3	8 - 4	5 - 0	25·36	2200	3	3	12
Barbour	Ramage & Ferguson	253	33 - 3	20 - 4	13 - 0	18·02	3800	3	1	{ 58 / 70
. Furness	Stephen & Sons	198	28 - 7	18 - 6		15	1400	3	1	9
. Newnes	Swan & Hunter	270	34			15	1800	3	1	
Mundy	... Fairfield ...	245	27 - 6	16 - 3		14·5	1250	2	1	5
King Edward	A. & J. Inglis	310				18	4000	3	1	
hedive of Egypt	A. & J. Inglis (rebuilding)	400	42	26 - 6		18	6500	3	1	
atham Railway Co.	... Denny Bros. ...	310	40	25	14 - 0	21·73	8500	3	1	{ 480 / 500
uth-Coast Rly. Co.	... ,, ...	280	34	22	9 - 0	21·5	6000	3	1	{ 480 / 510
and Larne Service	... ,, ...	300	40	24 - 6	10 - 6	20·7	6500	3	1	6
d Railway Co.	... ,, ...	330	42	25 - 6	10 - 6	22·3	7000	3	1	{ 670 / 730
,,	Vickers, Sons & Maxim	330	43	25 - 6	10 - 6	23·14	8500	3	1	{ 530 / 610
an S.S. Company	Armstrong, Whitworth & Co.	350	42	17 - 3	10 - 6	23 53	9500	3	1	4
atham Railway Co.	... Denny Bros. ...	310	40	25	11 - 0	22·9	9000	3	1	4
outh-Coast Rly. Co.	... Fairfield ...	280	34 - 8	14 - 6	9 - 3	21·75	6500	3	1	6
& J. Burns	... ,, ...							3	1	
stern Railway Co.	J. Brown & Co., and Laird & Co.	350	40		14 - 0	23	9500	3	1	4
n Government	... Cockerill ...	350	40		9 7	24	12000	3	1	4
Heligoland S.S. Co.	... Vulcan Co. ...	300	38		9 - 10	20	6000	2	1	
dia S.S. Company	.. Denny Bros. ...	275	44	25 - 6		18	6000	3	1	
Co. of New Zealand	... ,, ...	300	43	25	12 - 6	20·2	6300	3	1	
e S.S. Company	Hawthorn, Leslie & Co.	260	33	20 - 9	9 - 6	19	3500	3	1	
Co. of New Zealand	... Denny Bros. ...	400	50	33 - 6		17·5		3	1	
	Workman & Clarke	300						3	1	
S.S. Company	,,	540	60	42 - 6	27 - 6	19·5	12000	3	1	
ard Company	John Brown & Co.	678	72	52	32	21	21000	3	1	
,,	J. Brown & Co., & Swan & Hunter	760	88		32	25	65000	4	1	
R.N. ...	Armstrong, Whitworth & Co.	360	40	—	14 - 6	{ 21·75 / 23·63 } ‡	{ 2800 / 14000 }	3	1	{ 4 / 2
rman Navy	... Vulcan Co. ...	341	43 - 3	—	16 - 6	22 ‡	10000	4	1	
U.S.N. ...	Bath Iron Works	420	46 - 8	—	16 - 9	24	16000	4	1	
,, ...	Fore River S. & E. Co.	120	46 - 8	—	16 - 9	24	16000			
R.N. ...	Portsmouth Dockyard			—		21	23000	4	1	
,, ...						24	28000	4	1	
ench Navy ...	Société des F. & C. Mediterranee					21	1800	2	Various	18
,, ,, ...										
,, ,, ...	... Yarrow ...	152 - 6	15 - 3	8 - 4	5 - 0	26·4	2200	3	Various	{ 1
,, ,, ...	... Normand ...	125	14			26·5	2200	3	1	
,, ,, ...	... ,, ...	125	14			26	2200			
rman Navy	... Schichau ...	200	23		8 - 0	28·3	6000			
ne Turbine Company		140	17		7 - 0	18	1800	2	1	

‡ Designed. † On trial.

NE STEAMERS—GENERAL DIMENSIONS AND DATA.

	Length	Beam	Depth	Draught	Speed	Equivalent I.H.P.	No. of Shafts.	Screws per Shaft.	R.P.M.	Boiler Pressure.	Displacement.	Propeller Diameter	
	Ft. Ins.	Ft. Ins.	Ft. Ins.	Ft. Ins.	Knots.					Lbs.	Tons.	Ft. Ins.	
	100	9	—	3	32	2000	3	3	2300	210	45	1 - 6	Only one screw, 28″ dia
...	250	30	10 - 6	6	20·48	3500	3	{ 1 Centre / 2 Wing }	{ 503 C. / 750 W. }	150	700	{ 4 : 9 / 4 : 4 }	Put in service July, 190
...	270	32	11 - 6	6 - 6	21·43	4400	3	1	{ 750 C. / 1090 W. }	150	900		Put in service July, 190
& Co.	210	21	12 - 9	6 - 9	36·58	13000	4	2	1180	240	390	3 - 4	Launched 6/9/99. Ran
n & Co.	223	20 - 6	13 - 6	7 - 3	30·2	10000	4	3	1050	240	450	2 - 9	Sank at sea in Septembe
& Co.	210	21	12 - 9	7 - 3	27·1	7000	4	1	890	240	440	4 - 0	Reciprocating cruising en
	220	23 - 6	14 - 3	8 - 3	26·2	7500	3	2	940	250	570	3 - 3	
& White	175				26	3600	3	1	1200	220	225	3 - 0	12 building.
, Hawthorn,	250				33	15000	3	1	700	220	800	6 - 0	5 building.
	320				36	28000	4	1	600	250	1500	7 - 0	Details under considerat
...	152 - 6	15 - 3	8 - 4	5 - 0	25·36	2200	3	3	1200	225	145		One 3′ 0″ screw now fitt
on	253	33 - 3	20 - 4	13 - 0	18·02	3800	3	1	{ 550 C. / 700 W. }	180	1400	{ + 4 : 8 / 4 : 0 }	Yacht measurement.
s	198	28 - 7	18 - 6		15	1400	3	1	900	150	900		
r	270	34			15	1800	3	1		150	1250		Thames yacht measurem
...	245	27 - 6	16 - 3		14·5	1250	2	1	550	160	782		Thames yacht measurem
	310				18	4000	3	1			2800		
lding)	400	42	26 - 6		18	6500	3	1		150	3100		In process of conversion
...	310	40	25	14 - 0	21·73	8500	3	1	{ 480 C. / 500 W. }	150		{ 6 : 0 / 5 : 7 }	Screws originally arrang
...	280	34	22	9 - 0	21·5	6000	3	1	{ 480 C. / 510 W. }	150	1200		
...	300	40	24 - 6	10 - 6	20·7	6500	3	1	600	150	1750	5 - 0	Bow rudder fitted
...	330	42	25 - 6	10 - 6	22·3	7000	3	1	{ 670 C. / 750 W. }	150	1950	5 - 0	
axim	330	43	25 - 6	10 - 6	23·14	8500	3	1	{ 530 C. / 610 W. }	200	2000	{ 6 : 2 / 5 : 7 }	
th & Co.	350	42	17 - 3	10 - 6	23 53	9500	3	1	430	160		6 - 6	
...	310	40	25	11 - 0	22·9	9000	3	1	440	160		6 - 6	Sister ship "Invicta."
...	280	34 - 8	14 - 6	9 - 3	21·75	6500	3	1	600	150	1360	5 - 0	See "Engineering," Aug
...							3	1					
aird & Co.	350	40		14 - 0	23	9500	3	1	430	160			3 building.
...	350	40		9 7	24	12000	3	1	490	150	1950		Astern speed 16·0 knots
...	300	38		9 - 10	20	6000	2	1			2000		Curtis' turbines.
...	275	44	25 - 6		18	6000	3	1		150	2170		Sister ships "Linka," "
...	300	43	25	12 - 6	20·2	6300	3	1	650	150	2400	5 - 3	
& Co.	260	33	20 - 9	9 - 6	19	3500	3	1	550	160	1100	4 - 1½	
...	400	50	33 - 6		17·5		3	1					
arke	300						3	1					Also "Virginian," built b
	540	60	42 - 6	27 - 6	19·5	12000	3	1	275	180	13000	8 - 3	
Co.	678	72	52	32	21	21000	3	1	185	210	30000	14 - 0	2 building.
n & Hunter	760	88		32	25	65000	4	1	165	210		17 - 3	See "Engineering," Nov
rth & Co.	360	40	—	14 - 6	{ 21·75 ‡ / 25·63 ‡ }	{ 9800 / 14000 }	3	1	{ 480 C. / 490 W. }	250	3000	6 - 6	
...	341	43 - 3	—	16 - 6	22 †	10000	4	1	650		3200		
rks	420	46 - 8	—	16 - 9	24	16000	4	1	500	250	3750	6 - 6	Curtis' Turbines.
E. Co.	420	46 - 8	—	16 - 9	24	16000				250	3750		
kyard			—		21	23000	4	1	300	250	18000	9 - 3	Designs still under consi
					24	28000	4	1		250			
editerranee					21	1800	2	Various	1800		92	Various	Rateau Turbines. See T
												"	"
...	152 - 6	15 - 3	8 - 4	5 - 0	26·4	2200	3	Various	{ 575 H. / 1800 T. }		140	"	"
...	125	14			26·5	2200	3	1		250	95		Brequet Turbines.
...	125	14			26	2200				250	95		Astern speed 16·7 knots.
...	200	23		8 - 0	28·3	6000			865		350	Various	Curtis' Turbines.
	140	17		7 - 0	18	1800	2	1	650	250		4 - 6	

‡ Designed. † On trial.

REMARKS.
screw, 28″ diameter, now fitted to each shaft.
rvice July, 1901.
rvice July, 1902. Very largely used for experimental trials.
1 6/9/99. Ran ashore, and lost during Naval Manœuvres in 1901. Trials made in 1900.
ea in September, 1901,
ting cruising engines on inner shafts, 7½″, 11″, and 16″ × 9″ stroke. 400 R.P.M. Launched 2/1902.
g.
g.
nder consideration.
screw now fitted to each shaft.
asurement.
acht measurement.
acht measurement. Only twin-screw Parsons' installation.
s of conversion from paddle engines to turbines. Vessel built in 1865 by Samuda.
iginally arranged as in "King Edward." 13 knots astern speed.
er fitted
p "Invicta."
gineering," August 18th, 1905.
g.
eed 16·0 knots; 415 R.P.M.
rbines.
ps "Linka," "Lunka," "Lama.
rginian," built by Stephen & Sons. Weight saved by adopting turbines, 400 tons. Passengers increased 60.
g.
gineering," November 18th, 1904.
urbines.
till under consideration.
urbines. See Transactions Institution of Naval Architects, 1904.
" " "
" " "
Turbines.
eed 16·7 knots.
urbines.

NOTE.—Also projected two vessels for Great Central Railway Co., two for Allan Steamship Co., two for the Metropolitan Steamship Co. (New York and Boston Service), and various foreign warships.

Passing through a turbine system, as for instance a row of vanes, without giving up useful work or without losing heat, as is supposed with abiabatic expansion, the increase in kinetic energy per pound of steam, represented by the quantity $\frac{V_s^2}{2g} - \frac{W_s^2}{2g}$, equals the work due to the "heat drop" represented by $778\ (\lambda_1 - \lambda_2)$, in which $(\lambda_1 - \lambda_2)$ is the available heat in British thermal units between chosen pressures p_1 and p_2. The equation for steam speeds at exit of the vanes will therefore be

$$V_s = \sqrt{W_s^2 + 2g \times 778 \times B.\ T.\ U.}$$

The formulæ given by Mr. Speakman gives the exit steam speed in a turbine system where the entrance velocity to the vanes is *nil.* This may possibly be the case in the first row of guide vanes of a Parsons turbine (although a slight entrance velocity more probably exists even there), but this entrance velocity is certainly considerable in all the others. Pheripheral and steam speeds (V_t and V_s) being somewhat arbitrary, and being invariably settled upon by rules established from practical considerations, the formulæ may be used to determine the heat units absorbed by each row of vanes. As this primarily forms the basis for calculating the total number of rows (by dividing the total available heat with the individual B.T.U. "heat drop" of each row of vanes), which, in a measure determine outside dimensions and therefore weight, it is thought the suggestion may be usefully observed.

NOTES ON FIRING OF BOILERS ON RECENT TRIAL TRIPS.

By Lieutenant R. K. Crank, Member.

On the recent trial of the U. S. S. *Tennessee* the method of firing the boilers was as follows, viz :

There are sixteen Babcock & Wilcox boilers in the ship and each boiler has six furnace doors. For the trial these furnace doors were numbered, for any two boilers facing each other, as follows :

Forward boiler.

6 4 2 5 3 1

Fire room.

5 3 1 6 4 2

After boiler.

The signal to work the fires was given every minute and on the minute by a time keeper in each fireroom, who called out every minute the number of the furnace to be raked and coaled. The fires in each boiler were worked in succession, in the order in which they were numbered, *i. e.*, 1-2-3-4-5-6, one door in each boiler being opened when the signal was given. Each fire was first raked and then immediately coaled.

On the standardization runs each furnace door was kept open for nearly one minute (in some cases for a full minute) after the signal to work the fire had been given, and, as the signals were given at intervals of one minute, there was one door in each boiler open nearly all the time, or sixteen

furnace doors in all open nearly all the time, sixteen streams of air, each 12 inches by 16 inches, under a pressure of 2 inches of water, continuously rushing into the furnaces. A considerable portion of this delay in closing the furnace doors was due to the fact that the fireman had first to rake his fire, then get his shovel, then coal the fire, the door remaining open while he was getting his shovel. There were two firemen to each boiler. In the preliminary four-hour trial the two leading men on boilers O and Q opened and closed the doors for the firemen of these boilers; this greatly reduced the total time the doors were kept open, as the following data obtained from these boilers on the trial will show, viz:

	Boiler O.	Boiler Q.
Time taken to rake each fire, mean, seconds.....	19	19
Time taken to coal each fire, means, seconds.....	10	14
Interval between raking and coaling, seconds.....	4	5
Total time each door was open, at each signal, seconds..	34	38
Number shovels coal put on each time.....	5.3	4.2

Thus the time of keeping the doors open for these two boilers was reduced, on an average, from 60 seconds to 36 seconds, by having men to open and close the doors and by closing the doors while the fireman was getting his shovel.

At first, on the standardization runs, the fires were carried at a thickness of from 8 inches to 10 inches with an air pressure of about two inches; but as the results were not good, the thickness of the fires was increased to from 12 to 14 inches. On the preliminary trial the fires were carried at a thickness of from 12 inches to 16 inches. As better results were gotten with the thicker fires and 1.5 inches air pressure than with the thin fires and 2 inches air pressure, and the gas analysis showed better combustion, it is probable that the quality of the coal demanded thick fires. The coal was in part Pocahontas, in part New River, run-of-mine, with about 40 per cent. of slack. If a simple instrument could be devised for getting a quick measure of the CO_2 and O in the gases of combustion, it seems that this would be the surest and most reliable way of determining the thickness of fire and air

pressure necessary to get the best results from any particular coal. Mr. Peabody, of the Babcock & Wilcox Company, informed the writer that he had devised a rough-and-ready instrument for this purpore, an instrument that could be used to advantage aboard ship with B. & W. boilers.

With the thin fires and two inches of air the flame passed through only to the beginning of the second pass across the tubes. With the better firing on the preliminary trial the flame went through to the beginning of the third pass.

The system used on the trial of the U. S. S. *Louisiana* seemed to the writer to be better than that used in the case of the *Tennessee*. The Babcock & Wilcox boilers of the *Louisiana*, of which there are twelve, have four furnace doors each. For the trial, these doors, for a pair of boilers facing each other, were numbered as follows, viz:

Forward Boiler.

1	3	4	2
Coal	Coal	Rake	Rake

Fire Room.

Rake	Rake	Coal	Coal
2	4	3	1

The cycle of operations for any one boiler, beginning with furnace number 1, was as shown by the words under each furnace number in the diagram above, *i. e.*, 1st, coal No. 1; 2d, rake No. 2; 3d, coal No. 3; 4th, rake No. 4, the processes alternating. Then, after finishing with furnace No. 4, the cycle would be started again with the raking of No. 1, coaling 2, raking 3, coaling 4. This differs from the *Tennessee* method in that the operations of coaling and raking are performed alternately for any one fire.

The signals on the *Louisiana* were given by an electric signaling device, devised and made by the contractors and

automatic in operation. At intervals of about forty seconds this automatic device would ring a large gong and light a cluster of lights in each fireroom; this interval could be varied at will.

Each fire was coaled or raked every four minutes. The fires were carried at a thickness of from 6 to 7 inches, and the working of the fires proceeded in an orderly, uniform and admirable manner. As in the *Tennessee*, however, the doors were in many cases kept open too long by slow firemen.

These notes are written in the hope that they may contain some practical and useful suggestion as to the best manner of firing when running at full power.

HIGH-SPEED VEDETTE PINNACES.

By G. Simpson, Associate.

At the present time, when the problem of the selection of the future battleship is dividing the opinion of naval authorities, it is interesting to turn aside and consider that useful little auxiliary known as the vedette pinnace, more particularly as this Navy has not yet decided upon the most suitable type to adopt.

With a few exceptions these vessels are constructed of wood and are of about uniform length in the principal navies, although the speed and armament vary considerably. It might be well, then, to consider what is probably the best example of this class of vessel possessed by any of the powers and consider what improvements could be introduced to make them more efficient. The vessel under review is one of a number built for the Imperial Japanese Navy from designs prepared by the author. In arriving at this type various stages in evolution had been passed, so that up till now these boats have been considered as combining all the best features of exisiting types. The British Navy had adopted a boat of similar length with a speed of only 14 knots, while the Italians had adopted a 56-foot pinnace with a speed of 17 knots, and the Japanese themselves were contented with a like speed. About the time the battleships *Mikasa* and *Asahi* were built their admiralty considered that an advance should be made in this class of pinnace in the direction of increased speed, and it was to meet this demand on their part that the author undertook the design of these little vessels, and by reducing scantlings from what had been the practice in the heavier British vessels, rearranging the members to get the greatest strength on a minimum of material, and giving the easiest form known from a long experience in designing high-speed craft, he built the vedettes here reproduced, and, as far as is known, they have given entire satisfaction in service.

The principal dimensions are—

Length between perpendiculars, feet and inches	56- 0
Breadth, extreme, feet and inches	9- 4½
Depth, molded, feet and inches	5- 2
Displacement, tons	14.45

They were built on the two-skin diagonal system of teakwood planking, the inner thickness being ¼ inch, in strakes of 6 inches broad, laid at an angle of 45 degrees with the waterline, and the outer $\frac{5}{16}$ inch by 4 inches wide planked longitudinally. Between these skins insertion was tacked, consisting of thin calico soaked in lead paint, ensuring absolute watertightness. The outer planking was lightly caulked with boat cotton and the seams puttied. The deck was likewise laid in two thicknesses ranging fore and aft, with the seams and under strakes covered by the top planks and treated like the shell for watertightness. The stem was of ram form to increase the waterline length when "squatting" at full speed, and was worked from white oak grown to form, incorporating with an American elm keel by a "snipe's bill" scarph. The keel, made from a wood too much neglected for certain parts of boatbuilding in this its indigenous country, had the top sides and corners planed off to house the garboard ends of the diagonal planking, and a rabbet cut below this for the garboard strake. On top of the keel a suitable hog piece was worked to form a backbone and take the fastenings of the garboards. This hog runs all fore and aft, being swelled out and through-bolted to keel where the stern tube passes outside the vessel. To this member are clench-bolted the gunmetal sling plates by which the vessel is hoisted on deck. The top member consists of an American elm gunwale steamed and bent to form, into which are rabetted the two thicknesses of skin and deck. Timbers are worked about 10-inch centers of Canada elm splines steamed and bent to form, through-clench fastened with copper nails on rooves.

The transverse watertight bulkheads are of 2½ pounds galvanized steel plating flanged vertically to form stiffening, and connected to shell by a steel angle frame. In wake of

machinery spaces the beams are of galvanized steel angles, with extra strong beams in wake of conning tower. A glacis formed 2 pounds steel casing encloses the engine and boiler rooms, and is pierced with decklights and airports on sides. Forward is a 15-pounds steel conning tower enclosing the forward steering position and mounting a 3-pounder rapid-fire gun. A serving scuttle is fitted in deck immediately abaft the gun-working circle, through which the ammunition is passed from the magazine at the after end of forecastle. The forecastle is quite commodious for so small a boat, and is capable of berthing and sleeping in hammocks about half a dozen men. Right admidships are installed on each side a pair of torpedo tongs for side drop gear suitable for manipulating 14-inch Whitehead torpedoes. These, with the 3-pounder rapid-fire gun previously mentioned, comprise the armament, an offensive equipment not to be despised. The after part of the vessel is set apart for officers' use, consisting of a large open cockpit and a dwarf deckhouse well lit with square windows having hinging flaps on top. Some of the earlier boats have a rifle-caliber machine gun mounted on top of this house.

The boat is maneuvered by double rudders as shown, the invention of an American named Colonel Mallory. The great advantage of this system is found when maneuvering in going astern, a condition in which most vessels are defective with the ordinary single rudder. The rudders are connected with yokes and crossheads, and the actuating lever is the ordinary bar tiller sometimes used with a composition sliding gunter compensating the leverage at various angles of helm.

MACHINERY.

The propelling machinery, a general arrangement of which is here shown, comprises a single set of compound surface-condensing engines with cylinders 8½ and 18 inches in diameter with a 9-inch stroke, to which steam is supplied by a water-tube boiler of a working pressure of 175 pounds per square inch on an air pressure of 1¾ inches. The propeller is

of gunmetal, of the usual three-bladed type, 3 feet 9 inches in diameter and 3 feet and 10¾ inches in pitch, having a developed surface of 673 square inches. On trial, with a load of 3,500 pounds, these craft attained a mean speed of 19.25 knots and indicated 315 horsepower. To develop this power, however, they had to blow 3¼ inches of air, but when it is considered that the displacement was 13.75 tons the result is considered exceptional, showing a particularly easily driven form as witnessed by the High Admiralty co-efficient, namely 130. The revolutions per minute were 594, and the slip 12.95 per cent.

DESIGNS FOR A FIFTY-FOOT PINNACE.

The writer is of the opinion that all of the salient points of these Japanese vessels could be obtained on a length of 50 feet with the adoption of the Curtis marine turbine for propelling purposes, as the saving in space and weight would enable an armament of equal offensiveness to be carried, while at the same time the speed could be considerably increased and at least an equal radius of action obtained on the same bunker capacity. At the same time galvanized steel has been substituted in construction, as it is believed that this material has a great advantage over wood for boats of this class, being of equal weight for the same strength, in addition to which it is fire proof, and does not splinter, two points of first importance on a war vessel.

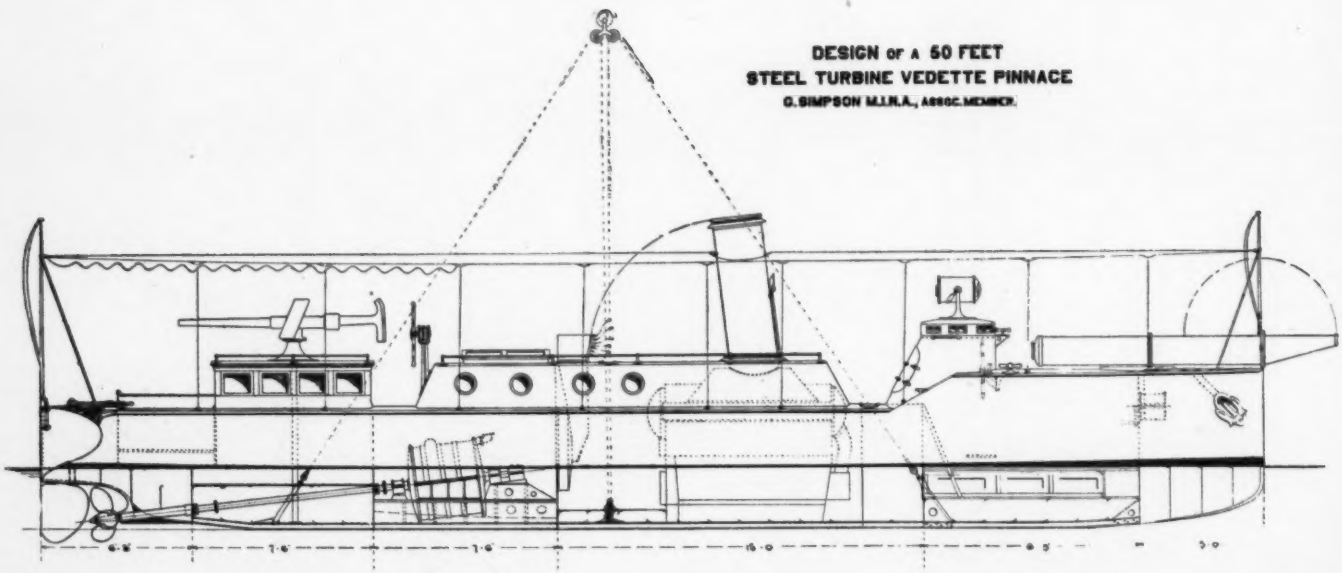
The design shown on plate has been prepared to show what to the writer appears the tendency of future construction in this line, and shows a steel pinnace of the following dimensions:

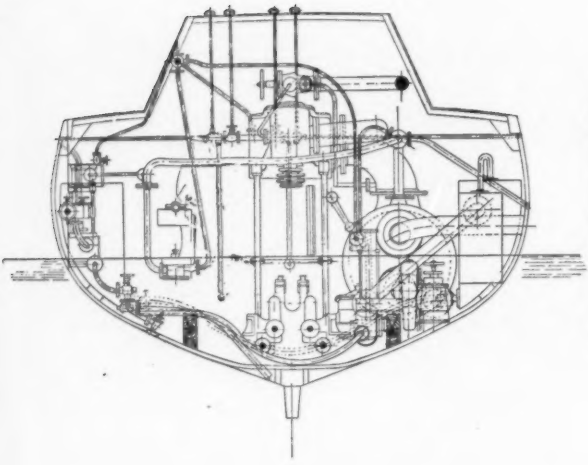
Length, extreme, feet and inches	50- 0
Breadth, extreme, feet and inches	8- 7
Depth, molded, feet and inches	4- 11
Draught, molded, feet and inches	2- 6
Displacement, tons	12
Slinging weight, tons	11

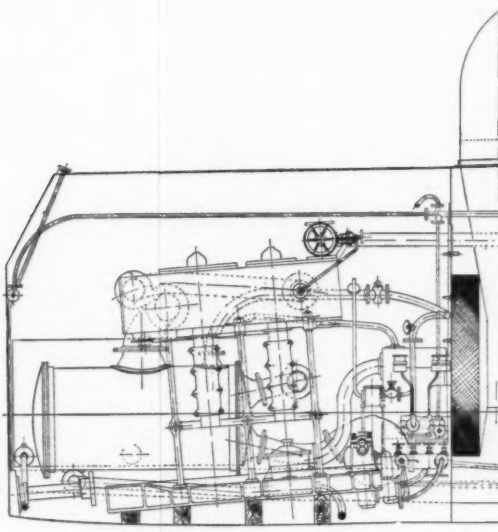
The shell, of 2½ pound galvanized steel, is riveted to ⅛ by ⅛ by 7 pound angle frames spaced 15 inches heel to heel, and the

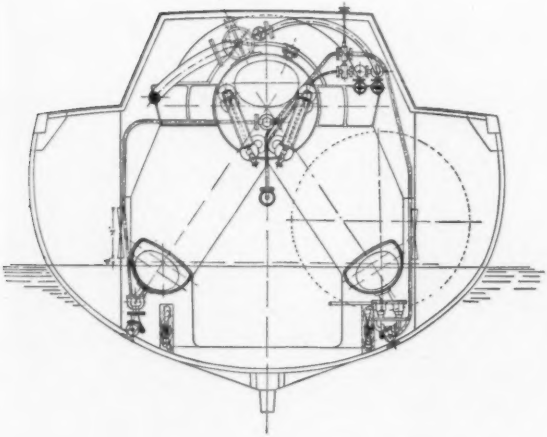
DESIGN OF A 50 FEET
STEEL TURBINE VEDETTE PINNACE

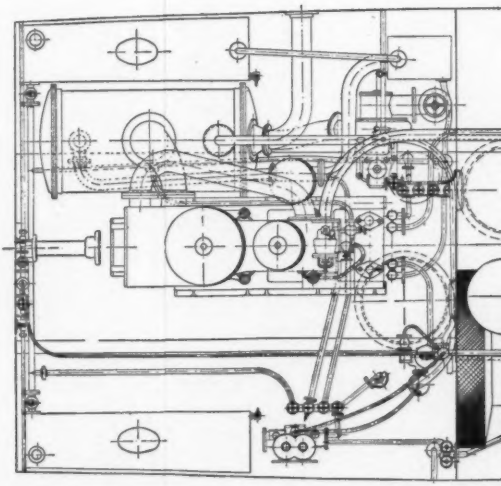
G. SIMPSON M.I.N.A., ASSOC. MEMBER.

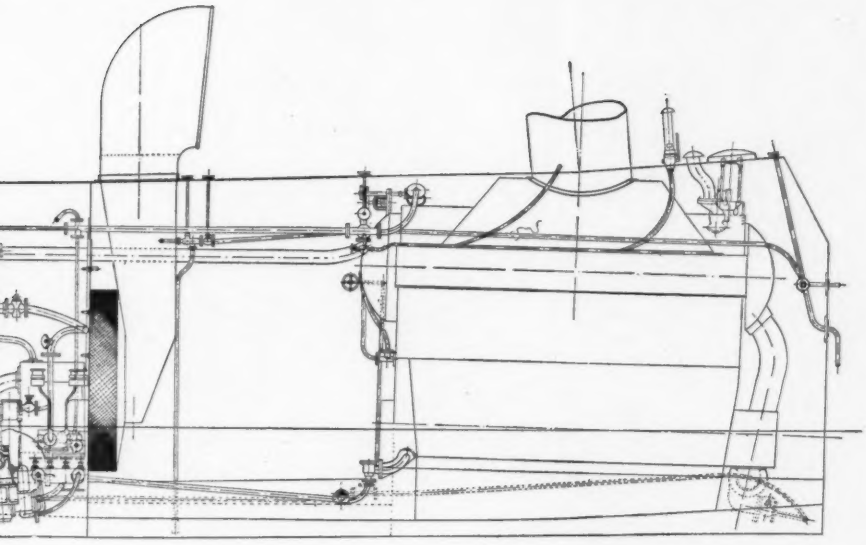


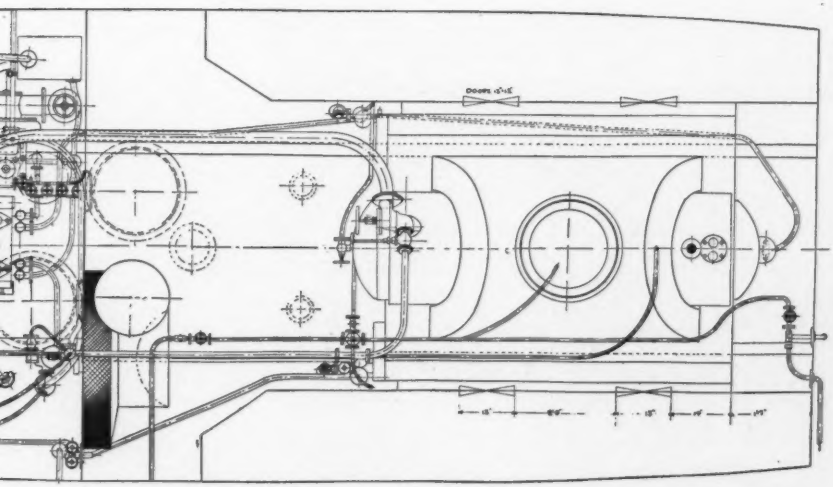






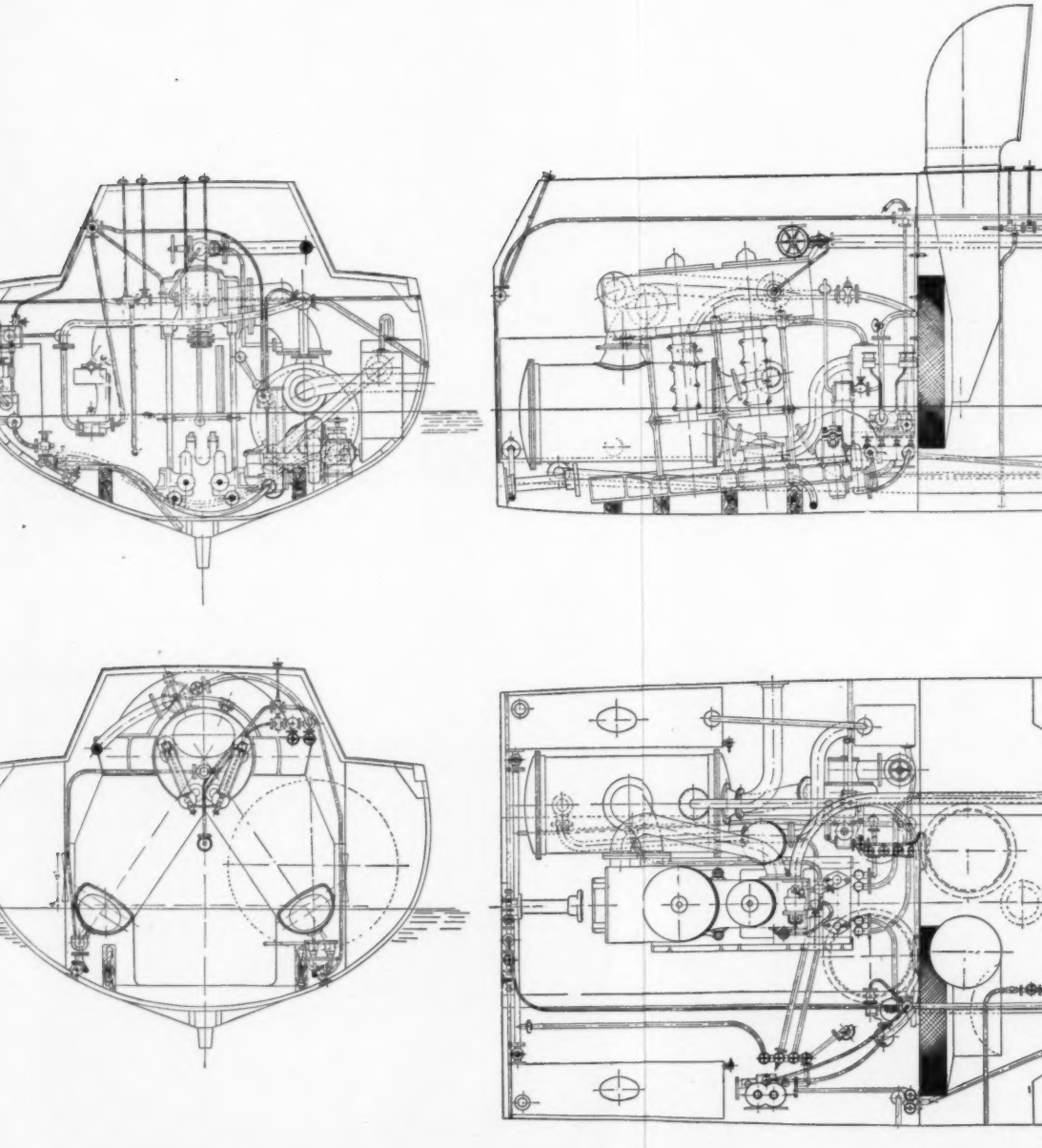


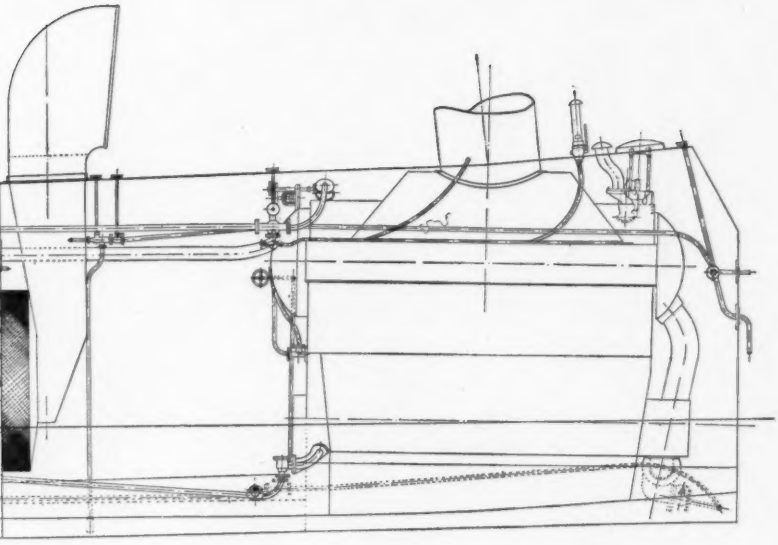


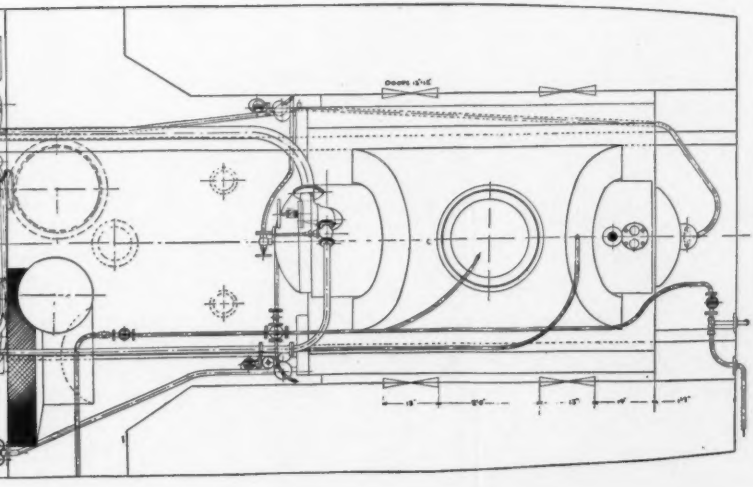


ARRANGEMENT OF MA
56 FT. VEDETTE

CYLINDERS	9½" and
STROKE	9 inches
R.P.M.	630
STEAM PRESSURE	180 lbs
I.H.P.	300
PROPELLERS	3'-9" Dia.
Pitch	3'-10½"
Dia.	4'3" Dia.
SLIP %	15







ARRANGEMENT OF MACHINERY
56 FT. VEDETTE

CYLINDERS	8½ AND 15 DIAMS.
STROKE	9 INCHES
R.P.M.	834
STEAM PRESSURE	190 LBS.
I.H.P.	300
PROPELLERS	3'-1" DIA.
Ds.	3'-10½ PITCH
Da.	573" DEVELOPED SURFACE
SLIP %	13

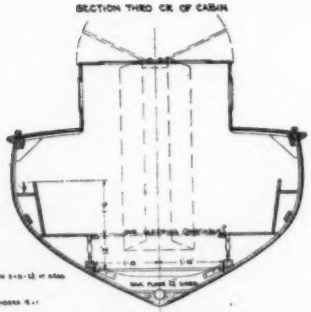
HIGH SPEED VEDETTE PINNA

AS SUPPLIED TO

THE IMPERIAL JAPANESE NA

DESIGNED BY

G. SIMPSON M.I.N.A., ASSOC. MEMBER.



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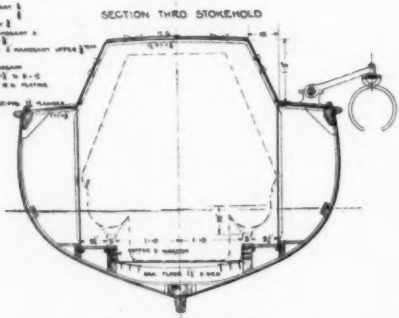
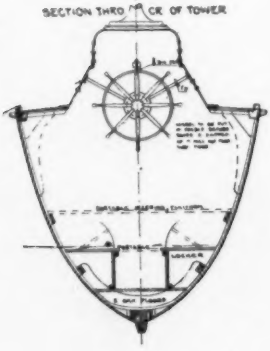
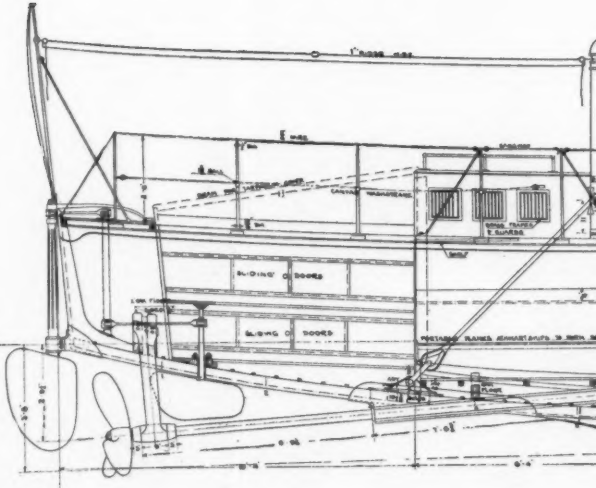
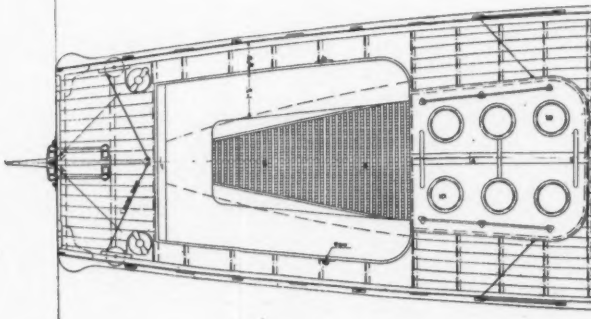








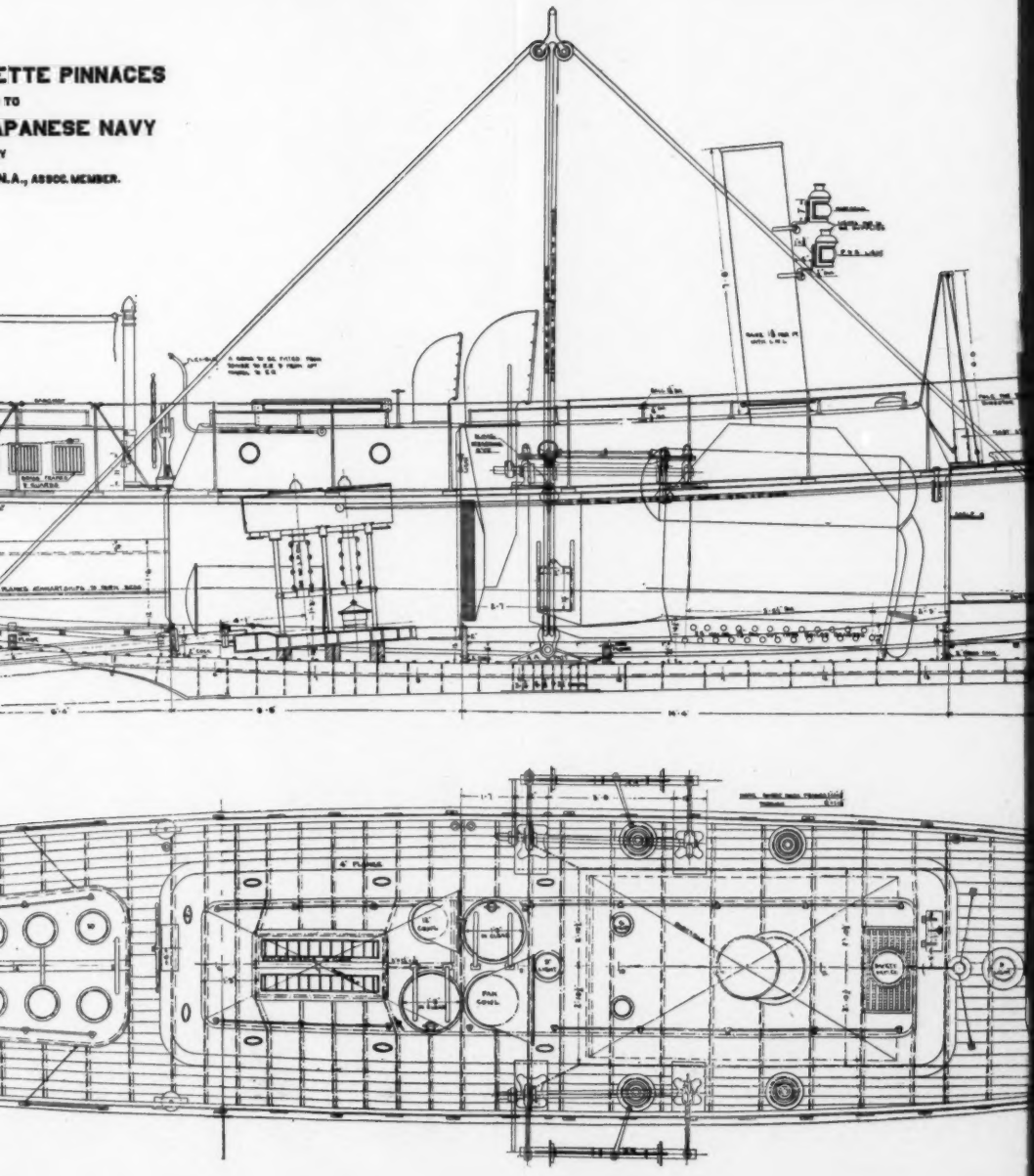
CTTE PINNACES

TO

APANESE NAVY

Y

N.A., ASSOC. MEMBER.



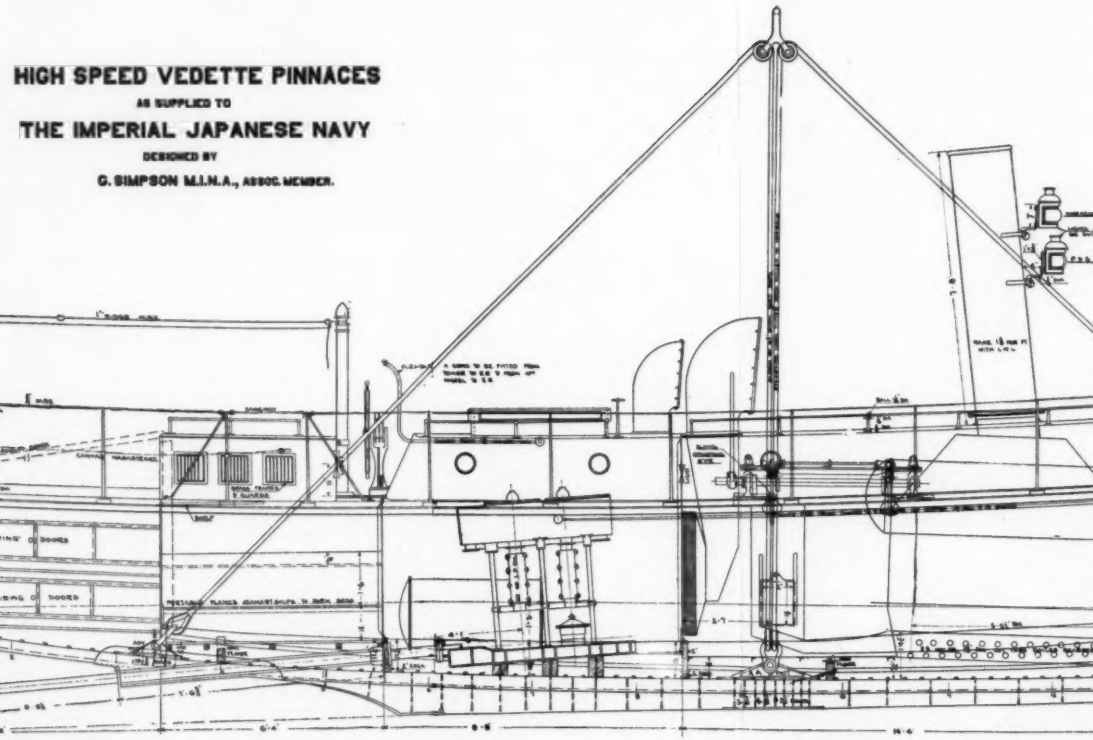
HIGH SPEED VEDETTE PINNACES

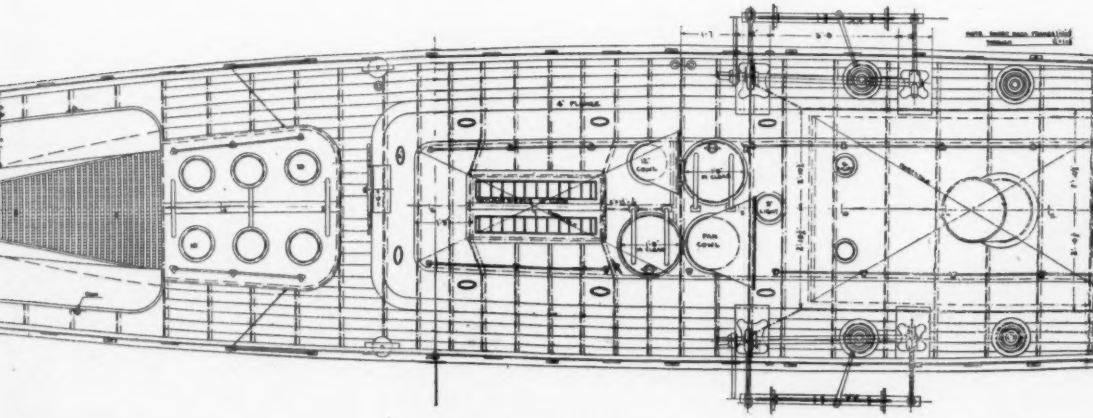
AS SUPPLIED TO

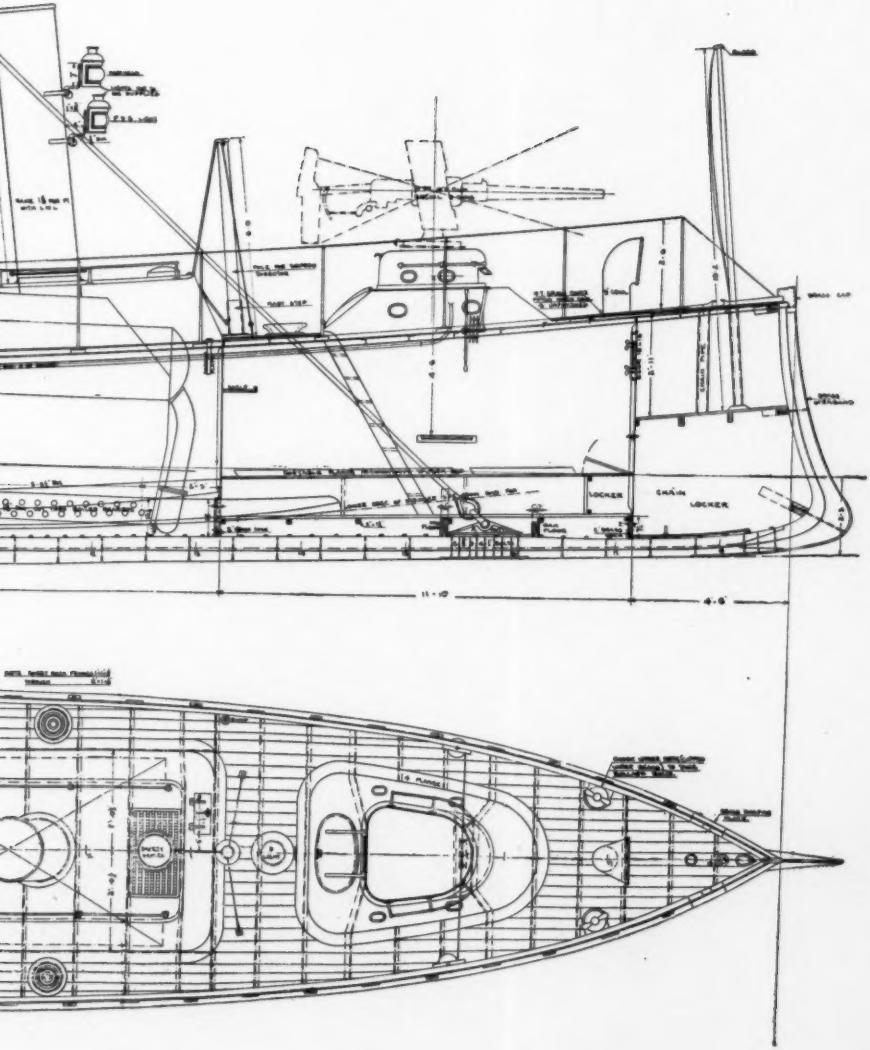
THE IMPERIAL JAPANESE NAVY

DESIGNED BY

C. SIMPSON M.I.N.A., ASSOC. MEMBER.






LOCKER
CHAIN LOCKER

backbone is formed with a 3-inch I section having reversed angle floors on top, giving the necessary strength for slinging and sling plates.

The hull is subdivided by five transverse watertight bulkheads built of 2 pound steel, plated vertically and flanged 1¼ inches to form stiffeners. The longitudinal strength is carefully provided for by fore-and-aft bunker walls throughout the boiler room, these being connected at the bottom to the shell and riveted to the casing coaming at top, forming a longitudinal girder of great strength. This plating is 2 pound suitably stiffened and braced. The deck plating is edge lapped with inside edges flanged ⅞ inch for stiffening, the whole being covered with linoleum ⅛ inch thick.

The arrangement of the boat follows generally the existing types with the exception of the forward part, which resembles the torpedo boat, the increased freeboard providing a better steering platform in dirty weather and for manipulating the searchlight, in addition to which roomier quarters are possible for the crew. A spacious cockpit and officers' cabin occupy the after part of the vessel.

The armament comprises one rifle-caliber machine gun mounted on top of cabin and a 14-inch torpedo tube placed on raised deck forward, permitting of a head or bow fire, the tube mouthpiece being hingeable to allow of stowing it inboard on top of tube. A conning tower of 15 pound nickel-steel is built at after end of forecastle, on which is installed the torpedo director, and inside of tower the forward steering station is located. On top of the conning tower may be placed a 3-pounder rapid-fire gun when torpedo tube is unshipped, or alternatively a 16-inch electric-light projector.

The propelling machinery consists of a 36-inch two-stage Curtis marine-type turbine supplied by a Fore River patent water-tube boiler with a working pressure of 220 pounds per square inch when blowing 2 inches of air and driving the turbines at 700 revolutions per minute. The speed of the vessel with a 2,240-pound load under the foregoing conditions will be 20 knots per hour.

THE BURSTING OF METAL CHAMBERS UNDER INTERNAL AIR PRESSURE.

BY LIEUTENANT COMMANDER L. H. CHANDLER, U. S. N., MEMBER.

Compiled from Records of the Bureau of Ordnance by permission of Chief of Bureau of Ordnance, Rear Admiral N. E. MASON, U. S. Navy.

Some years ago the Bureau of Ordnance received reports that the English Navy was in possession of a steel to be used in forged torpedo air flasks which, when charged to full working pressure, would withstand dropping from a height of ten feet, or the impact of the projectiles up to and including the one-pounder, without explosive disruption. It was not meant, of course, that the flasks would keep out the shell (which was a sand-loaded dummy), but merely that the shell would penetrate, allowing the air to escape through a small hole thus made without causing an explosion of the whole flask.

A series of experiments in regard to this matter, covering several years, was accordingly carried on at the Torpedo Station by direction of the Bureau of Ordnance, and it is the object of this paper to give some account of these experiments, of their results, and of the conclusions drawn from them.

NO. 1 EXPERIMENT—BALLISTIC TEST.

Air Flask.—The flask used in this experiment was one originally intended for use in the construction of a steel, 5-meter, 45-centimeter, mark I torpedo, but which was rejected because

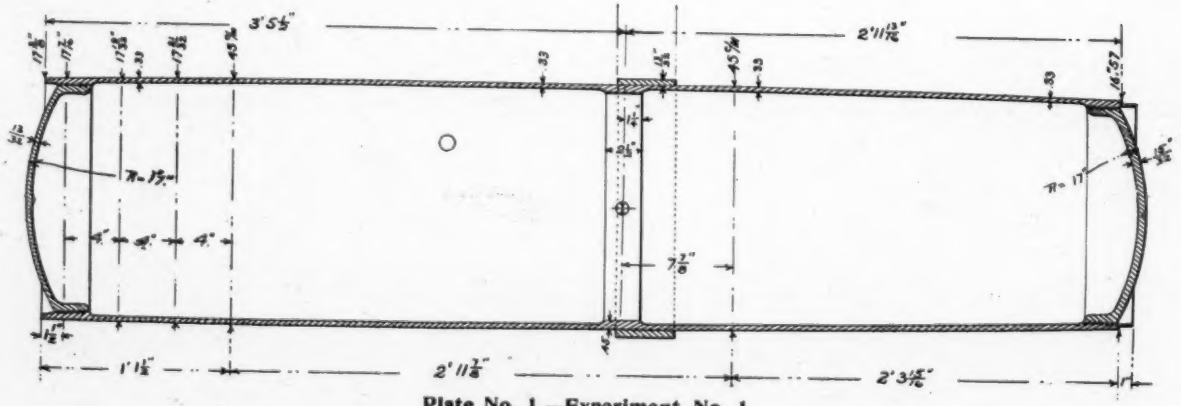
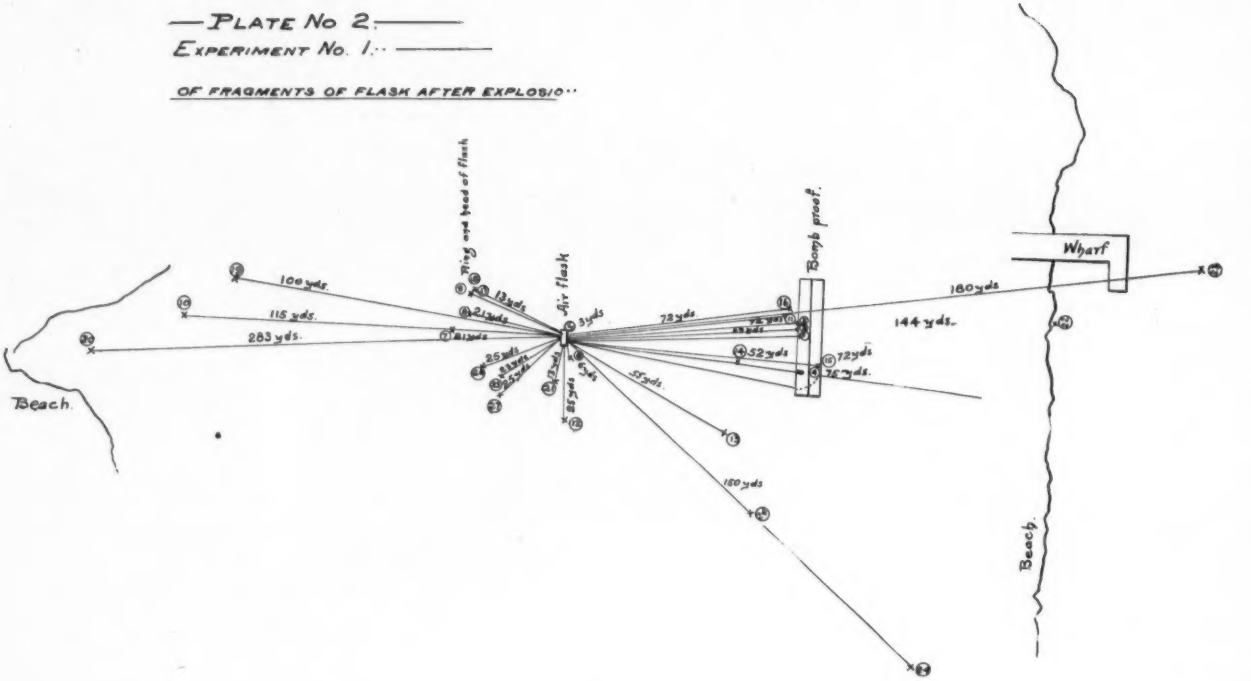


Plate No. 1.—Experiment No. 1.

— PLATE No 2. —

EXPERIMENT No. 1. —

OF FRAGMENTS OF FLASH AFTER EXPLOSION



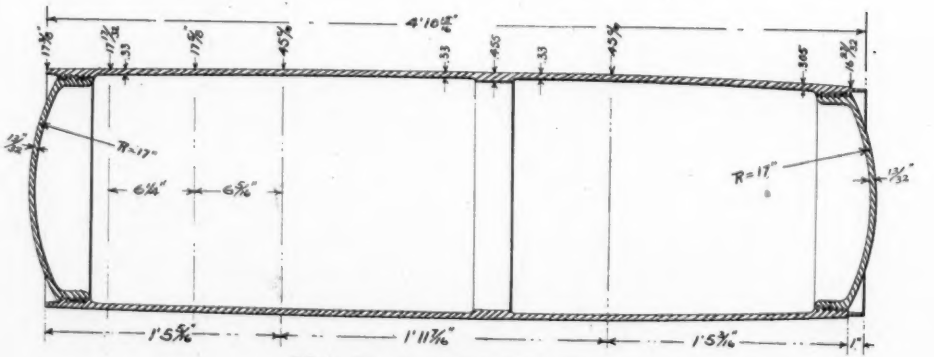


Plate No. 3.—Experiment No. 2.

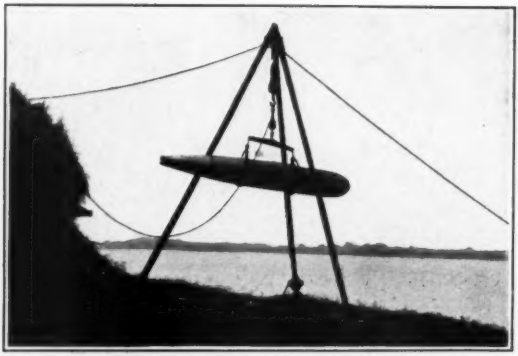


Plate No. 4.—Experiment No. 2.

Plate No. 5.—Experiment No. 3.

Plate No. 6.—Experiment No. 3.

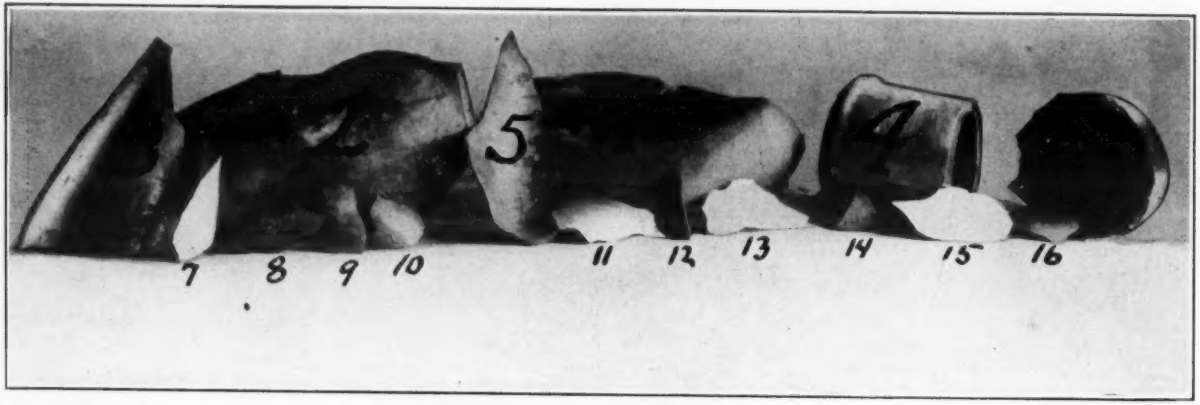
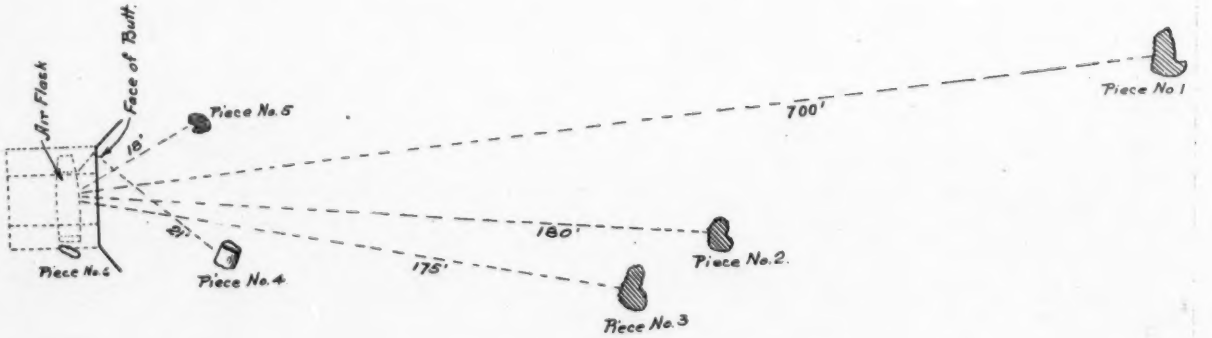


Plate No. 7.—Experiment No. 3.

PLATE No. 8.

EXPERIMENT No. 3.

DISPERSION OF FRAGMENTS OF FLASH AFTER EXPLOSION



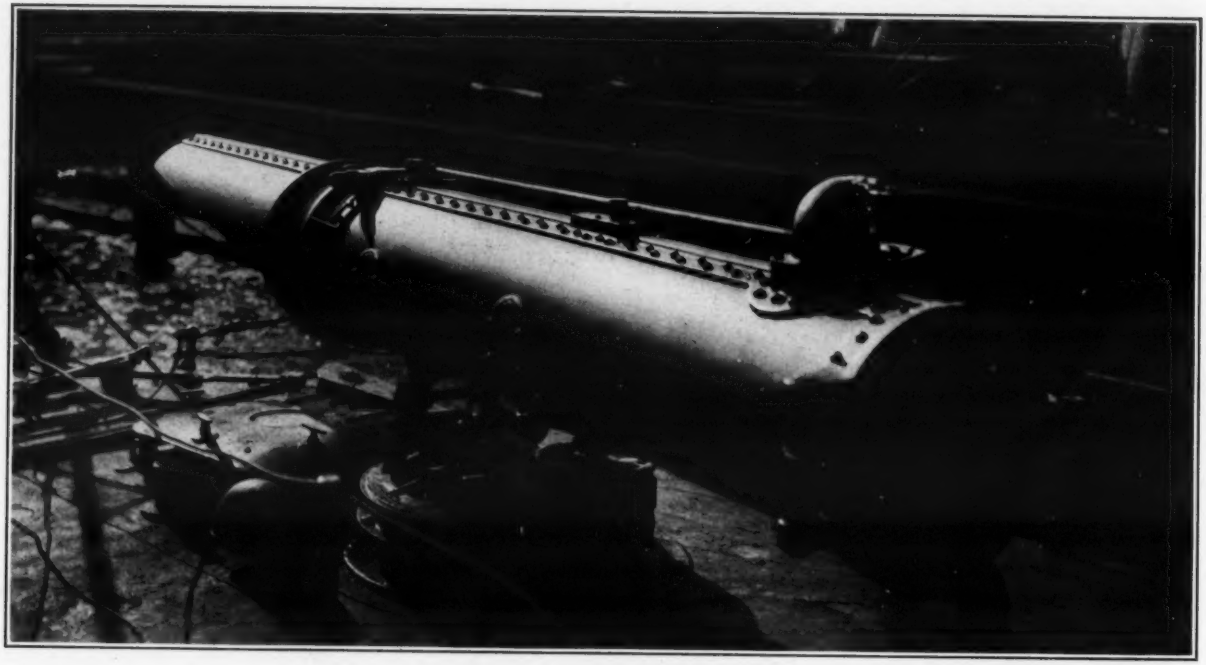


Plate No. 9.—Experiment No. 4.

LAUNCHING TUBE FOR 3.55 METER. 45 CENTIMETER TORPEDO.

Plate No. 10.—Experiment No. 4.

Plate No. 11.—Experiment No. 4.

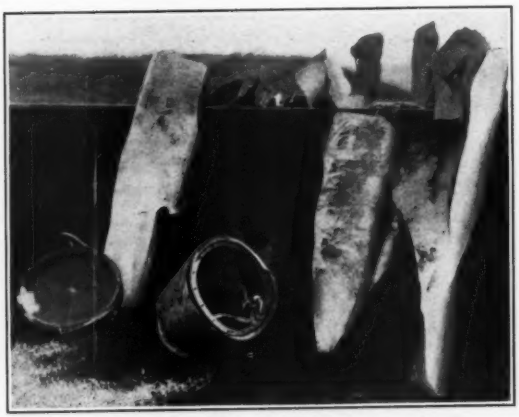


Plate No. 12.—Experiment No. 4.

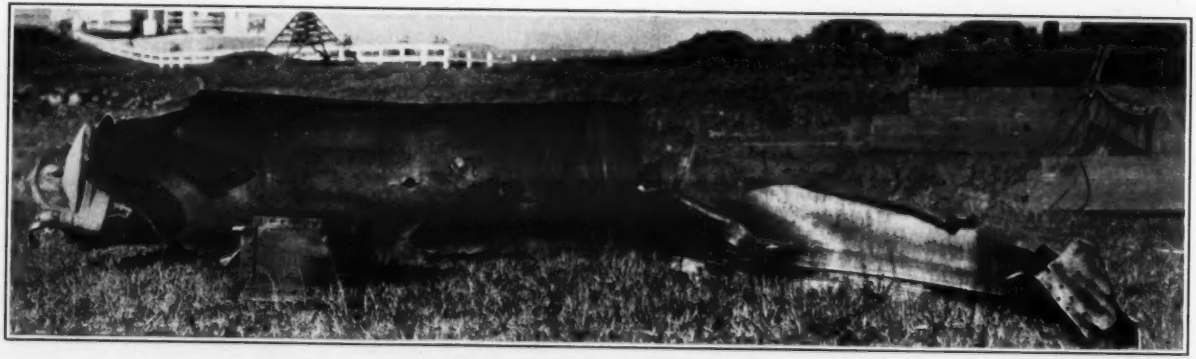


Plate No. 13.—Experiment No. 4.

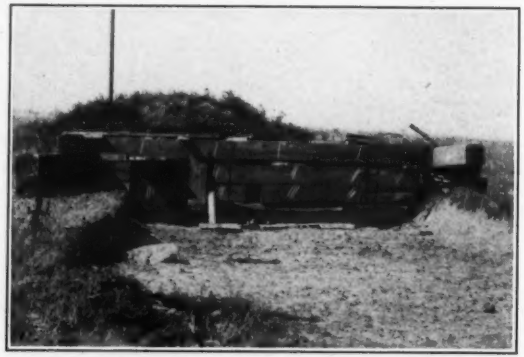


Plate No. 14.—Experiment No. 5.

Plate No. 15.—Experiment No. 5.

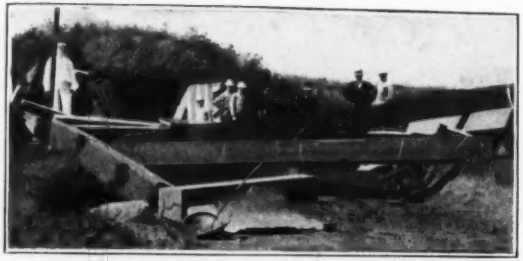


Plate No. 16.—Experiment No. 5.

Plate No. 17.—Experiment No. 5.

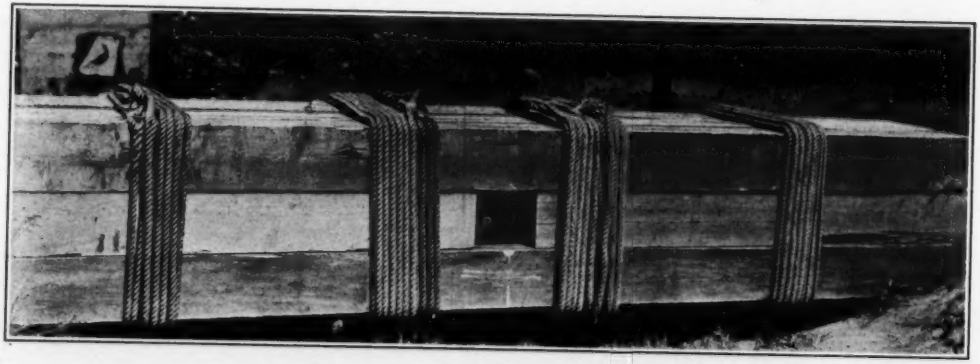


Plate No. 18.—Experiments Nos. 6 and 7.

Plate No. 19.—Experiment No. 6.

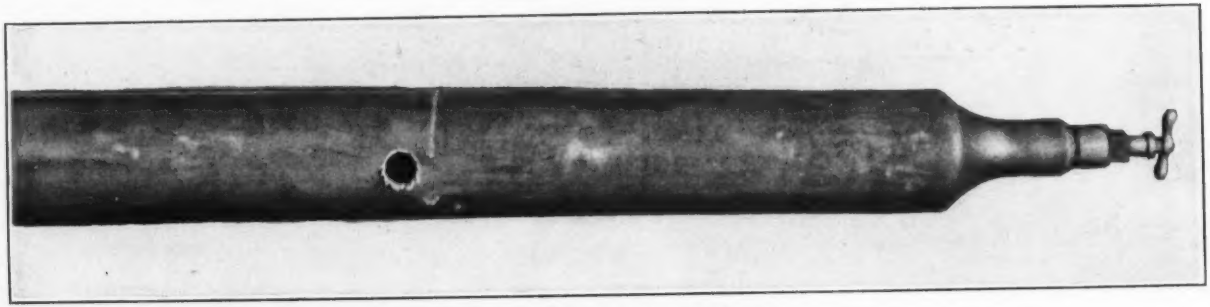


Plate No. 20.—Experiment No. 6.

of a small defect. A plain forged-iron band was shrunk on around the defect, as shown in the accompanying plate (No. 1), and the resultant flask was thought to approximate very closely, for purposes of ballistic test, to the actual strength of a sound flask as actually furnished with the torpedoes.

The dimensions of the flask were:

Outside diameter at forward end, $17\frac{7}{16}$ inches.
Outside diameter at rear end, 16.57 inches.
Maximum outside diameter, 45 cm.
Flask, cylindrical for part of its length, and then tapering to each end.
Normal thickness of flask, .33 inch.
Flask thickened on inside in wake of guide stud to .45 inch for 2.5 inches of its length.

Characteristics of metal of flask:

Elastic limit, 68,000.
Elongation in 2 inches, 14 per cent.

The band shrunk on to cover the defect was $\frac{13}{32}$ of an inch thick and $4\frac{1}{32}$ inches wide, and was of ordinary forged iron, its forward edge being 3 feet $5\frac{1}{8}$ inches from the forward end of the flask.

The flask was given the usual hydrostatic test of 2,025 pounds per square inch internal pressure, and under this test showed an elongation of 0.1144 inch and a dilation of 0.026 inch. The normal expansion under the same type of flask under this test was 0.1084 inch in elongation, and 0.03 in dilation.

The normal air pressure used in these flasks in service is 1,350 pounds per square inch.

Elements of Firing.—The flask was charged to 1,350 pounds per square inch air pressure, and set up in the open, with its axis perpendicular to the line of fire. The gun used was a 6-pdr., using 400 grammes of smokeless powder, and with its muzzle placed seventy yards from the flask. This

Plate No. 20.—Experiment No. 6.

would correspond to a striking velocity of about 2,300 foot seconds.

Results.—The flask burst explosively upon impact, pieces being thrown as shown by Plate No. 2. The distances to which the larger pieces were thrown are shown by the following table, in which the numbers of the pieces correspond to those on the plate.

No. of fragment.	Weight in pounds.	Distance thrown, in yards.	No. of fragment.	Weight in pounds.	Distance thrown, in yards.
1	1	69	15	1	72
2	1	69	16	2	72
3	1	69	17	68	13
4	1	75	18	37	13
5	2	3	19	50	100
6	13	6	20	5½	283
7	29	21	21	3	13
8	14	21	22	40	144
9	7	13	23	Lost	180
10	3	115	24	23	150
11	3	72	25	6	23
12	4	25	26	4	25
13	3	55	27	2	25
14	4	52			

It should be noted from the plate that the pieces were generally thrown in directions approximately perpendicular to the axis of the flask.

NO. 2 EXPERIMENT—DROPPING TEST.

Air Flask.—The flask used in this experiment was that of steel torpedo, 3.55-meter, 45-centimeter, mark I, No. 66, of which the general dimensions are shown in Plate No. 3. The characteristics of the steel in this flask were:

Elastic limit, 65,000.

Elongation in 2 inches, 12 per cent.

Nature of Test.—This test being to determine the resistance of the flask to dropping, it was fitted with a coil and a head, and was thus up to the full weight of the torpedo in service (845 pounds). It was charged to the full service air charge of 1,350 pounds per square inch pressure.

Test.—At a safe place sheers were erected and the torpedo hoisted until the bottom of the flask was six feet above the vertical edge of a ⊥ angle iron placed transversely to the axis of the torpedo and resting on some two-inch boards firmly supported on solid rock. The torpedo was dropped electrically.

Results.—On the first drop some difficulty was experienced with the detaching apparatus, and when the torpedo fell it did not strike a fair blow on the vertical or web edge, but struck the edge of a flange, which was pointed up at about forty-five degrees from the vertical. There was no apparent damage to the flask, but the flange of the angle iron was turned over by the blow. The angle iron was strengthened and the test repeated.

The second test was successful. In it the torpedo, weighing 845 pounds, with the flask charged to 1,350 pounds per square inch air pressure, was dropped six feet vertically, the bottom of the flask striking squarely at the middle point of its length, across the vertical edge of the ⊥ angle iron, the top edge of which had been fined down to one-eighth of an inch in thickness. The angle iron was neither bent nor dented by the blow. The flask remained perfectly intact and apparently fit for use, the dent in it being so slight as to almost escape detection; it was a mere scratch, from one-sixty-fourth to one-thirty-second of an inch deep at the deepest point.

Plate No. 4 illustrates this experiment.

NO. 8 EXPERIMENT—BALLISTIC TEST.

Air Flask.—The same flask was used as in the preceding experiment, charged to the same pressure. Further details

in regard to this flask are that the flask itself had been tested to 2,025 pounds hydraulic pressure, at which it gave an elongation of 0.0973 inch and a dilation of 0.021 inch. From its acceptance, on December 17, 1892, it had been used for purposes of instruction at the Torpedo Station.

Conditions of Test.—The flask, without either head or tail, was placed in a firing butt, well under its roof, resting unsupported at the back of its chamber upon a box.

Gun.—A 1-pdr. gun was used, firing a dummy shell with smokeless powder. This gave a striking velocity of approximately 1,996 foot-seconds.

Result of Test.—The flask burst explosively upon impact, with damage to the butt as shown in Plates Nos. 5 and 6. Some of the timbers of the box upon which the flask rested were slightly damaged. The distribution of fragments was as shown in Plates Nos. 7 and 8 and the following table:

No. of Fragment.	Weight in inches.	Distance thrown in Test.
1	55½	700
2	71½	180
3	34	75
4	84	21 (see below).
5	21-½	18 (see below).
6	65	(see below).

Fragment No. 4 was the tail end of the flask, with immersion chamber, which first struck one side of the butt near the roof, and was then thrown across the line of fire to a distance of 21 feet.

Fragment No. 5 first struck the roof of the butt, and was then thrown 18 feet to the front.

Fragment No. 6 was the head end of the flask, and was found embedded in the timbers of the butt.

The nine small pieces shown in Plate No. 7 were found scattered in front of the butt at distances from 10 to 180 feet.

The small pieces aggregate in weight about 60 pounds. The total weight of the fifteen fragments was about 390½ pounds. The weight of the original flask was 389 pounds.

NO. 4 EXPERIMENT—BALLISTIC TEST.

Air Flask.—The flask used was an exact duplicate of that used in experiments Nos. 2 and 3, except that it had received a small dent in 1898, as the result of a collision between a merchant steamer and the *Columbia.* It was otherwise in good condition, and stood the usual hydrostatic test of 2,025 pounds without abnormal results just prior to the experiment. It was charged for the experiment to full service air pressure of 1,350 pounds.

Conditions of Test.—In order to determine the effect of the explosion of an air flask aboard ship, this flask had the exercise head and after body attached, and it was placed in a standard launching tube (Plate No. 9) built of plain steel ⅜-inch thick. The tube was then enclosed in a timber baulk, as shown in Plate No. 10, the common axis of the torpedo and tube being at right angles to the line of fire. The timbers used in constructing the baulk were of spruce, 12 inches × 12 inches × 16 feet, the whole structure being lashed together with manila and wire rope. A small square opening was left to admit the projectile.

Gun.—A 1-pdr. gun was used; charge 67½ grammes of smokeless powder; projectile, a dummy 1-pdr. armor-piercing shell; distance from muzzle to flask, 160 feet; striking velocity, about 2,200 foot seconds.

Results.—The projectile struck the tube and flask normally on the center line. The resultant explosion was very violent, as shown in Plate No. 11, the effect on the timbers being probably increased by the extra lashings and fastenings. The force of the surrounding envelope of timbers was mostly upward and to the rear. The timbers nearest the gun (forward face of structure) were thrown about 20 feet towards the

gun and were practically uninjured. Of the timbers over the tube, one was thrown up in the air and very badly splintered, and the other two were thrown up into the air from 50 to 75 feet, falling from 30 to 40 feet to the rear. The rear timbers were thrown to the rear, several being broken and much splintered, one piece, about four feet long, being found 100 yards to the rear and right. The lashings, both wire and rope, were entirely demolished, pieces of the wire rope being found from 100 to 200 feet away. The end timbers, those which protected the ends of the tube, were intact; those on the left side being still in place, while those on the right were simply thrown down. The pieces of the flask and tube were scattered among the wreckage, but were held within a radius of 20 feet by the surrounding timbers.

The flask was broken into five large and eight small pieces, as shown in Plate No. 12. The two heads were intact, and each with a circular band of the flask surrounding it all around and covering the screw threads, the rupture taking place a short distance from the heads, where the middle of the flask proper sheared all the way round. The impact of the projectile on entrance into the flask was well marked and regular, with the usual characteristics of a shot hole in steel. The exit hole was not as well defined, being somewhat irregular, with radiating cracks. There were no signs of corrosion inside the flask, which was found to be well coated with thick, gummed oil.

The launching tube was ruptured into four large and four small fragments, the rivets along the top seam having been sheared the whole length, and a strip about one foot wide, which contained the guide slot, was torn out longitudinally and entirely separated from the rest of the tube. The spoon joint was blown off, and the joint band was blown off and torn to pieces. The ingress and egress holes of the shell through the tube were well marked. The fragments of the tube are shown in Plate No. 13.

The shell was found in the firing butt, unbroken, but badly deformed, with head mushroomed and band gone.

NO. 6 EXPERIMENT—BALLISTIC TEST.

Air Flask.—The flask used in this test was of bronze, of approximately the following composition:

Copper, ... 92.02 per cent.
Tin, ... 8.05 per cent.
Specific gravity, ... 8.80.

No data as to physical characteristics are available. The flask was about 17 inches in diameter.

Conditions of Test.—The location, gun charge, striking velocity, etc., were the same as in experiment No. 5. There being no tube available, however, the charged flask was surrounded by timbers without the connecting tube, but the timbers were not as securely lashed and fastened as in the preceding test, it being thought that such taut lashings only added force to the explosion; the only object of the timbers being to eliminate danger from the flying pieces, and to render the recovery of the fragments possible. (Plate 14.) The flask was charged to 1,350 pounds per square inch air pressure.

Results.—The flask burst explosively upon impact, as shown in Plate No. 15. The projectile struck the flask normally on the midships line, punctnred both sides of the flask and one of the back timbers, and was lost. The general effect of the explosion on the surrounding timbers was not as great as in the preceding experiment, perhaps partially owing to the smaller capacity of the bronze flask (about two-thirds that of the steel), and partly to the fact that the surrounding timbers were not as tightly secured. The explosive effect this time was to the front and up, instead of to the rear and up, as was the case with the steel flask.

On explosion the top timbers were thrown from 15 to 20 feet into the air. The front timbers were thrown from 20 to 25 feet to the front, toward the gun, one piece being badly broken up, a fragment of which, about 3 feet long and split

from end to end, being thrown about 50 feet toward the gun. Two of the rear timbers were thrown about 10 to 15 feet to the rear, but were unbroken. The timbers across the ends of the flask were left practically in place, but thrown down. (Plate 16.)

The flask was torn and ruptured into three pieces, one head being sheared entirely off with the flask metal still attached by its screw threads, and the other head with a piece of the flask metal partly bent away and separated, but still attached in one place by the solder used in screwing in the head. The inlet and egress holes in the flask were true, with all the usual characteristics of shot holes in bronze, the metal around the egress hole showing several small radiating cracks. The interior surface of the flask was polished and clean, and had evidently been smooth finished before the heads were placed. (Plate 17.)

NO. 6 EXPERIMENT—BALLISTIC TEST.

Air Flask.—The flask used in this test was not that of a torpedo, as was the case in each of the preceding tests, but was one of the elements of the latest type of air accumulator for the torpedo plants of large ships. It was a cold-drawn, bottle-necked cylinder, closed at one end, 51½ inches long, 6¾ inches in outside diameter, and 5 5/16 inches in inside diameter.

Conditions of Test.—The flask was lashed up inside of what was practically a wooden box, whose sides, top and bottom were each 12 inches thick, all being wedged and frapped together with lashings. (Plate No. 18.)

Gun.—A Krag-Jorgensen 30-caliber rifle ball was the projectile employed, the flask being 160 feet from the muzzle of the gun, the charge being 35 grains of smokeless powder, the muzzle velocity about 2,000 foot seconds and the bullet having the ordinary cupro-nickel jacket.

Results.—The bullet simply perforated the flask and re-

mained inside. There was no explosion, the air which had been at 2,250 pounds per square inch pressure, simply escaping through the hole. The punctured flask is shown in Plate No. 19.

NO. 7 EXPERIMENT—BALLISTIC TEST.

Air Flask.—The flask and the air pressure were the same as in the preceding test.

Conditions of Test.—These were also the same as in the preceding test.

Gun.—A 1-pdr. gun was used in this test, firing a one-pound dummy armor-piercing shell. The charge was 67 grammes of smokeless powder, the distance of the muzzle of the gun from the flask was 160 feet, and the muzzle velocity was about 2,100 feet.

Results.—There was no explosion. The shell passed completely through the flask, and was lost in the butt. The holes in the flask were fairly cleanly cut, and the air escaped through them very rapidly with no explosive effect. The punctured flask is shown in Plate No. 20.

GENERAL REMARKS ON ABOVE TESTS.

It is to be noted that no gun cotton or other explosive was involved in these tests, the explosive results following impact being solely due to the high-pressure air charge.

In this connection it is worthy of mention that repeated experiments have been made by firing dummy projectiles into masses of wet gun cotton, and it has been found that an explosion almost never follows such impact. To explode the war head of a torpedo it would probably be necessary for the projectile to strike the detonating apparatus directly.

As a result of the first five experiments described, it was decided that the torpedo air flask, when charged, constituted a real danger, much greater than did the gun cotton in the war head. This fact had much to do with the earlier decision that torpedoes should only be installed aboard heavy

ships when they could be amply protected by armor, and with the later decision to only install them aboard such ships for discharging from submerged tubes. Of course, aboard torpedo vessels the danger of the explosion of a flask is one of the legitimate risks of that type of craft.

Following the torpedo-flask experiments has come the replacing of the old steel accumulators, which would also undoubtedly explode from impact, by the bronze ones described in experiments 6 and 7, which are apparently not explosive. The greater weight of this bronze necessary to ensure a given strength has so far made it difficult to employ it for torpedo flasks, as the cutting down of weight is a most important feature of torpedo design.

SUGGESTIONS FOR THE CARE AND OPERATION OF NAVAL MACHINERY IN THE ENGINEER DEPARTMENT U. S. NAVY.

By Lieutenant H. C. Dinger, U. S. N., Member.

These notes are the beginning of a series of articles to be published in the Journal of the American Society of Naval Engineers. They are designed to describe in a simple and detailed manner the main features in connection with the care and operation of naval machinery and have been gathered from various sources that have been available. Discussion is invited, with a view of making the suggestions as valuable as possible for those in charge of Naval Machinery. —Ed.

Getting Underway.

BOILERS.

The time required preparing to get underway depends upon the type and size of the boilers in the vessel, as well as upon the character of the engines.

Ships having large Scotch boilers will require six to eight hours at least, twelve hours is better. Vessels with water-tube boilers may require less than an hour for the boilers, but it is difficult properly to warm large engines in so short a time.

The boilers to be used and the numbers of them should be selected with due consideration for the location of coal supply, and the condition of all the boilers in regard to repairs, state of cleanliness, etc.

If the boilers are empty, they must be pumped up to slightly below steaming level; if kept full of water, water is run down; the usual practice being to pump the water in to some double

bottom or reserve tank with auxiliary feed pump, and if no special system of piping for this purpose is fitted, use fire-room hose.

The water line at starting fires should be several inches below working level, especially in water-tube boilers. On starting up there may also be considerable water which has collected in condenser or feed tank; to prevent the necessity of its being run into bilge, this water should be pumped to boiler.

Make sure that all broken joints about the boilers are re-made, and that everything is tight; that all surface and bottom-blow valves of boilers not in use are closed.

Prime fires by placing a thin layer of coal on grates (not done till water is at steaming level). Start fires, generally by using live coals from boilers in use.

Keep ash-pit doors closed at first and open furnace doors to allow air to reach the fire above the fuel. In Scotch boilers, start water circulating apparatus, if any be fitted, as soon as fires are lighted. If there is no water circulating apparatus, a circulation may be set up by putting auxiliary pump on the bottom blow of boiler and discharging into feed line. The general type of circulator in Scotch boilers is the hydrokineter where the circulation is produced by a steam jet. By means of this device the water in boilers may be kept at 150 to 180 degrees Fahrenheit, using steam of auxiliary boiler, and under these conditions steam can be got up in a very much shorter time without injuring the boilers. With hydrokineters in use steam has been raised in very large boilers in less than two hours.

If there is no boiler in operation the fire is started with kindling wood or oily waste, laying this near front and topping it with small lumps of coal, which will take fire as the wood burns. Adding a little oil to the coal will cause it to catch fire more quickly.

The fire is allowed to burn up gradually and is pushed back as it burns, new coal being added from time to time till full thickness, 6 to 10 inches, is reached.

After fires have been started, the fire doors are closed and ash pit doors opened.

As soon as a slight pressure is shown on gauge, open air cock to allow air to escape. Other practice is to allow all air to pass into steam line and engines during warming up boilers, tap valves being opened for this purpose, but it may not be desirable to start to warm up engine till some time after steam has been formed in the boilers.

With Scotch boilers great precautions must be taken that the boiler is gradually heated and that one part is not cold while another is hot.

Steam in Scotch boilers should form in about five or six hours after fires are started.

In water-tube boilers fires can be pushed as much as possible, and a great many precautions that have to be observed with Scotch boilers can in a measure be disregarded. In emergency, steam can be gotten up to working pressure in water-tube boilers in 20 to 30 minutes, but it is advisable to allow at least one hour.

When steam instead of air comes out of air cock, close it. In Scotch boilers, there should be about two hours from time steam begins to form till working pressure is reached. In water-tube boilers it may be only a few minutes.

ENGINE ROOM.

About an hour before the time set for getting underway, start to warm up engines. The length of time for this depends on the size of engines. For very large engines more than an hour should be allowed; for small engines an hour is not necessary.

TO GET READY.

See all tools, material, clothing, etc., clear of engine.

See all parts of engine in place and properly secured.

Disconnect jacking gear.

Start circulating pumps, making sure that injection and discharge valves are properly opened.

If vessel has been lying in tropical parts where there are a large number of jelly fish, the injection strainers should be blown out with steam.

Start main air pumps and open main exhaust valves to condensers if valves are fitted.

Open bleeder to drain steam line, open other drains in steam line. On large vessels, and especially where there are pockets in the main steam line, as on *Tennessee* and *Washington*, great care must be exercised that steam line is properly drained.

Open bulkhead stops.

Drain separator.

The draining of steam pipe is very important, for if it is not done there is great danger of water hammer and a rupture of some steam pipe.

Open boiler stop valves, unless they have been open since fires were started, just cracking them open at first, and then open them gradually. This should at first be done with boiler that has steam up, the other boilers being connected as their pressure approaches the pressure of boilers that are connected.

Get reversing engine ready and turn steam on it.

Turn steam on jackets and drain them. Open cylinder drains.

See all working parts ready, oil pipes and cups clear and wicks and oiling gear at hand; take gaskets off bearings. Put a little oil by hand on the principal pins, slides and bearings, slack stern gland.

The steam being up to throttle, move links back and forth with reversing engine and crack open throttle slightly. This allows a little steam to pass into H.P. cylinder and warms it up. Crack open pass-over valves to allow steam to enter I.P. and L.P. cylinders.

Continue to work links back and forth, and thus get all parts gradually warmed up.

Turn steam on capstan and steering engine, when ordered from deck, and have pump ready for washing off anchor chain.

Try engine-room telegraph, also telegraph to fireroom, voice pipes, bells, telephones, etc.

See what feed pumps are in proper condition and try all that are to be used.

The boilers being connected and all main stops opened, see all oiling apparatus ready, including oil cans, oil syringes, graphite and cylinder oil for swabbing ronds. See reserve-feed tank full and oil cups filled. The boiler stop valves are never opened wide; usually 2½ to 3 turns of the wheel is sufficient.

An efficient aid to internal engine lubrication is to introduce graphite into cylinders just before starting. This can be easily done by mixing it with water and putting it down an indicator pipe if no lubricator cups are fitted.

Circulate water through thrust bearing and slides.

Make inspection to see that there is plenty of steam, good fires, good vacuum, and everything about engines ready and all men at stations.

Permission to try engines should be obtained about fifteen or twenty minutes before time set to be ready. Permission being granted, engines are tried.

TRYING ENGINE.

Before trying engine, pass order to stand clear. Then open throttle slightly and move links back and forth. The engine will then start one way or the other. If it does not start well, it is probably due to water in cylinders, which must be gradually worked out. A well handled and properly adjusted engine will start without manipulating pass-over valves.

Try engine both going ahead and astern and see that it reverses easily. Do not move engine more than a few revolutions, usually two, each way, as a greater amount of turning is likely to bring a strain on anchors or moorings.

After trying engines, take final look to ascertain if there is any heating of bearings or any indication of wrong adjustment; everything being ready, the department is reported

ready to get underway. After this report is made, a machinist must stand by to answer any signal promptly.

In trying engines, there may be several causes to prevent proper working; below are some of them.

Water in cylinders.—This will prevent the engine moving full stroke; the engine is likely to start, move a little way and then bring up. The water must be gotten rid of by opening all drains and moving piston back and forth by reversing gear till the engine will turn a complete revolution.

Engine on center.—This can be guarded against by taking care when engine is stopped that the H.P. crank is up near middle of stroke. Reversing engine, and bringing link back again, will throw most engines off the center. If this fails, pass-over valves can be used. In using pass-over valves, regard must be had for the position of cranks so that all efforts are to move the engine in the same direction. Sometimes an engine may fail to move on account of bad handling of the pass-over valves, whereby one cylinder is trying to move the engine one way and another in the opposite direction, the result being that the engine fails to move.

By opening pass-over too much at starting, in engines having attached air pumps and slide valves, a considerable pressure may be obtained in receivers, and hence on back of valves, if slide valves be fitted. This may be sufficient to prevent the moving of the links. The remedy is to close pass-overs and open drains so that pressure on backs of valves may be reduced.

This failure of reversing gear may also take place at any time engine is stopped during maneuvering when pass-over valves are left open or a considerable pressure on slide valves is produced by other means.

Whenever an engine fails to move with a slight opening of throttle, the cause of obstruction should be found, and the engine should not be forced by opening throttle wide, as by so doing something may be broken.

Valve rods sticking.—Valve rods sometimes stick; they may be loosened by applying oil, loosening stuffing box or loosen-

ing up on valve-stem guide. Some of the moving parts may be caught in some way, and at times the jacking gear has not been disconnected.

Excess water on starting.—With attached air pumps, in starting up, a great deal of water will be thrown into tank; in such cases it is advisable to put a pump on the air-pump channel ways and pump directly to boiler. This will prevent tank from overflowing and losing fresh water in bilge.

Handling, general.—Some engines handle much easier than others, and all have some peculiarities which are only found out by experience; but a great many general features hold for all. The real handling of an engine can only be learned by practical experience where practically all the senses are developed to apply to the case.

Turning over.—When engine is turned over in answer to signal, wicks should be put in and then special lookout must be had for heating or any maladjustment.

Throttle.—The throttle must not be suddenly opened or closed, but the engine, especially if a large one, should be allowed to work up to full power gradually.

Reversing.—When reversing, the throttle should first be closed, and engine then reversed; but when the emergency signal is received (that is, a signal to back full speed when going ahead), the engine should be immediately reversed, even at a risk of breaking something. This signal should *never* be given from deck unless it is an emergency signal.

RUNNING ENGINES UNDERWAY.

Whenever there is any doubt or trouble in connection with the operation of the engines, the officer of the deck should be notified immediately so that he can make suitable dispositions to avoid danger.

The most laborious work in connection with the running of engines will be when steaming in squadron, when exact distance must be maintained. With same steam and vacuum and no change in links, the pressure in the H.P. and I.P. receiver will be the same for the same number of revolutions,

so that, after some running, these guages can be used as a quick indication of the number of revolutions.

Electric revolution indicators are fitted on some vessels which indicate by a pointer at the end of every minute or half minute the exact number of revolutions the engine has turned during the preceding minute.

With well adjusted engines and ordinary care in operation, the main engines are not likely to give much trouble.

The points to look out for in the engine room underway are, in general:

1. A watchfulness and oversight where all the senses are used to detect any derangement or improper working.
2. Keep all moving parts properly lubricated.
3. Keep water out of cylinders.
4. Keep steam pressure, links and throttle so adjusted that engines work smoothly and economically and that there is the proper distribution of work between the cylinders.

GENERAL OVERSIGHT AND WATCH.

The senses that come into use in connection with the operation of engines are sight, hearing, touch, smell and taste.

The eye.—The trained eye is able to detect when various moving parts are not moving with regularity, as well as to notice movement in parts that should not be moving. The eye, of course, can detect such things as steam leaks, water leaks, cracks and breakage.

Hearing and touch.—A great deal concerning the proper running of an engine can be told by the sounds heard in the engine room. The various sounds tend to combine into a sort of rythm, and the experienced ear can tell when matters are going well and when even very slight deviations from this ordinary rythm are present, and this forms a very efficient means of detecting and locating faults.

The chief sound is the rather dull and heavy thump caused by the pounding of the main or crank-pin bearings. This can be located by observing which crank has just passed the center when the thump is heard, remembering that the thump will be

heard after and not before passing the center, or by filling bearing with oil the thump will be lessened. If the bearing is felt and found to be warm, or the thump felt by other means, the knock is located. The main bearing will have a duller sound than the crank pin or crosshead and will be felt all over the structure. The crosshead knock is a sharper sound and is not heard at so great a distance.

Often lost motion can be told by feeling the bearings. Placing a piece of metal against one part and the other end against the teeth or ear will also give indications.

A valve being loose on its stem or piston loose on rod causes a solid, sharp thump or a dull click. On starting up or slowing down, slide valves are apt to rattle or cause a clicking sound due to lack of pressure on their backs.

Attached air pumps, on becoming flooded, give a heavy thump due to excess of water; or this may be caused by the absence of an air cushion, which may be remedied by opening air cock on bonnet for a moment.

Spring rings vibrate more at certain speeds and cause a rasping sound at times. Other sounds inside the cylinder are caused by moisture in the steam and by the rings and cylinder walls becoming dry. When this rasping sound is heard while engines are in free route, a little cylinder oil put on piston rod or some graphite and water introduced into cylinder will relieve the noise. Scraping and grunting, caused by water in the piston rings, is generally only heard on starting up and slowing down.

The spring rings of piston valves give a sort of whistling sound when dry.

The cylinder relief valves are apt to rattle when there is little pressure on them. Stuffing boxes sometimes squeal, due to insufficient lubrication.

The churning sound noticed about engines is due to the passing of steam through ports. There is a difference in sound showing whether the port opening is ample or not. This difference may be noticed by running cut-off well in and out and observing the change.

Touch is of great use in feeling the degree of heat and to determine slight movements not otherwise noted.

The sense of *smell* is of great value in detecting heating or improper lubrication, as the smell of heated and burning oil can be noticed.

Taste comes into use in tasting feed water to see whether it is fresh or brackish.

HEATING.

Various moving parts are liable to heat. This is caused by (1) improper or insufficient lubrication, (2) bearings being out of line, (3) bearing set up too tight, (4) grit or dirt from inferior oil or from engine room getting into bearing, (5) uneven surface of bearing or brass.

Bearings must be frequently felt and special regard had for lubrication. If there is a sort of lather on the bearing, it is a sign of proper lubrication. Good oil runnng out shows that there is too much oil. Heating or the smell of burning oil shows a lack of lubrication.

If bearings get warm, the supply of oil should be examined and increased. If it gets warmer, water may be sprayed on or circulated through, according to which arrangement is supplied. But water should be avoided, if possible, the danger being that all parts are not cooled at the same time, and there is danger of bending, warping or cracking due to sudden and unequal contraction caused by the cooling effect of the water.

If the temperature of a bearing continues to rise after having increased the oil supply and we have a hot bearing, other means must be taken.

The danger of a hot bearing is that the brasses may expand till they grip the journal, thus causing a constant increase in the friction and heat.

With increased heat the white metal may run.

Slowing down will reduce the friction.

Slacking back on cap bolts will increase the clearance and prevent gripping; but these cannot be slacked back too much, as pounding will result.

By stopping and readjusting the heated bearing, the matter may be remedied. This is probably the best way, as it is the surest. If the bearing is not large, it can be taken off, cleaned out, covered with graphite and a liner inserted in a few minutes, and on starting off it will be all right. It is thus a very good practice to have paper and thin copper liners, already cut, at hand for all bearings, so that there is no delay in fitting them.

For cooling a bearing, tallow, black lead and sulphur are advisable as they absorb heat.

Water should not, except in most extreme cases, be sprayed on a hot bearing and should only be used before very great heat is manifested. A great objection to the use of water is that, being lighter than oil, it gravitates to the lower part of the bearing and thus displaces the oil, and after water is once started in a bearing, it is difficult to get it properly oiled. Water may be constantly used on guides and thrust bearings which are arranged hollow so that the water does not come on the sliding surfaces.

Often on trial trips streams of water are run continually on all the principal bearings, but in those cases the water is started before the heating begins. This is, however, not good practice for ordinary running.

Heating due to bearings being out of line or poor surfaces cannot be effectively remedied except by readjusting the parts, the only remedy in these cases being to run slow and to use oil freely.

OILING.

Oiling in connection with the operation of engines consists in, (1) keeping the oil boxes or cups supplied with oil, (2) supplementing this supply by hand where necessary, (3) oiling by hand small bearings that do not have separate pipes, (4) lubricating piston and valve rods, (5) internal lubrication, (6) regulate water service.

The oil boxes are usually filled from a reservoir tank located in top of engine room with pipes leading to all the various boxes so that they can be filled. Where no such system is provided, the boxes are filled by using an oil measure. Oiling by hand

is done with the squirt can; a small half-pint can is usually the handiest. In addition to the squirt can the oil syringe is used to reach parts that are out of the way, such as slides, piston rods, etc. For the fast moving engines, syringes are generally used in place of the squirt can.

Piston rods and valve rods are lubricated by means of swabs, using cylinder oil mixed with graphite, also by oil syringes.

On large bearings tallow cups are provided, and these have to be filled by hand.

The covers of all oil boxes and cups should be kept closed to prevent dirt from getting into oil.

There are no definite rules to be given regarding the frequency of lubricating. This can only be told by experience in each particular case.

Amount of oil required.—This will depend largely on the skill of the oiler, the adjustment of the bearings and their condition, the quality of the oil and the temperature of the engine room. In hot weather the oil will run freer and wicks should be decreased; in cool weather the oil will run slower and wicks should be increased.

In time wicks rot and also become clogged, thus changing the amount of oil supplied.

Fair practice is to use about one gallon of oil per ton of coal, or one gallon per 1,000 H.P. per hour. For small engines, such as torpedo-boat engines, considerably more oil per horsepower will be used than for large engines.

INTERNAL LUBRICATION.

For vertical engines no internal lubrication is necessary other than the swabbing of rods and wiping out of cylinders and vaselineing them at times when they are opened. The addition of a small quantity of graphite from time to time as cylinders give indications of becoming dry is advisable. When cylinders have once obtained a good surface by the use of graphite, they will run well without much further lubrication.

POUNDING.

Results from three general causes:

1. Too much clearance in a journal, slide or connection.
2. The use of too light oil on heavy pressure.
3. Improper distribution of power between the different cylinders.

The first is remedied by readjusting, and in part by slowing down or changing speed, as at certain speeds pounding is greater.

By the use of heavier oil the second cause may be removed, for them a heavier and thicker film of oil is provided.

The third cause is remedied by a readjustment of cut-off. Means for increasing compression will generally reduce pounding, but as a rule this can be determined from trial.

LEAKS.

The stuffing boxes as well as various joints may leak slightly. These leaks make themselves visible by the issue of steam or water. Steam joints are likely to leak slightly on starting up when there is considerable water in the pipes, but as they become hot and water is driven away they again become tight. Tightening up on the bolts of such joints slightly will often stop the leaks, provided the gasket is still good.

Stuffing boxes, when they leak, may usually be set up while running, but care must be taken that they are not set up too tight so that they do not produce excessive friction or cause the rods to become hot. It is better to have a little steam leak than to have a hot rod. With well fitted metallic packing there is little danger of either excessive leak or of great friction, and for all rods of any size metallic packing is fitted. It must be remembered that all steam leaks are a direct loss of power as well as of fresh water, and the loss by leakage sometimes amounts to 5 per cent. of the total steam used.

PRIMING, WATER IN CYLINDERS.

The presence of water in cylinders is dangerous, especially in quick moving engines having small clearance spaces. A

great many broken cylinder heads and other serious accidents have been due to this cause. The pressure of water is shown by a cracking or snapping noise when there is not very much, and by a heavy, sharp thump when there is considerable quantity The relief valves cannot very well take care of this because the pressure comes all of a sudden at the end of the stroke. When the presence of water is noticed, the drains should be opened. If this does not relieve the cylinder, close down throttle somewhat, thus wire drawing the steam more. If the water still increases, the throttle should be closed further, steam line drained, and the cause of the priming looked for at the boilers and there remedied.

Separator.—The separator must be kept drained, for if a large amount of water collects it may go over in a mass to the engines, thus defeating the very purpose for which the separator is fitted.

JACKETS.

Most large engines are fitted with steam jackets. These are very useful in warming up cylinders since they distribute the heat to all parts of the cylinder walls.

Most engines have all cylinders jacketed; others only have the I.P. and L.P. cylinder jacketed.

The office of the jackets is to prevent liquefaction in the cylinders, and they thus increase the economy of the engine, the jacket steam giving up its latent as well as its sensible heat to prevent the steam in the cylinders from condensing. Since none of the heat of the jacket steam is lost in the condenser, its apparent theoretical economy is very great.

Considering the action of the steam alone, the most efficient practice would be to carry full boiler pressure in each jacket, but due to mechanical considerations, such as strength of castings, etc., this is not done, and if jackets are used the steam pressure in each should be kept slightly higher than the pressure in the corresponding receivers in order that the jacket steam may be hotter than the steam in the cylinders.

With slow moving, medium pressure, long-stroke engines,

jackets undoubtedly result in economy, but with high-pressure, high-speed, short-stroke engines, there are various mechanical considerations that tend to reduce the advantages from their use. With that class of engines it has also been demonstrated by experience in some special cases that jacketing has little advantage in economy.

High-pressure jacket steam causes the cylinders to become hot and dry, prevents steam from being moist, and thus reduces the natural lubrication in the cylinders. This results in what is known as scrooping. When this is noticed the jacket pressure should be lowered. The steam being dry and likewise the rubbing surfaces, there is considerable friction, as well as wear, with attending difficulties, scoring of cylinders and breaking of rings, so that though jackets may have an economy, other troubles brought about by their use may at times be sufficient to make it advisable not to use them. As the greatest trouble due to friction and wear occur in the H.P. cylinder, where there is probably the least economy due to their use, it would appear that jackets should only be used on the I.P. and L.P. cylinders.

In cold weather it would seem to be more advisable to use jackets than in warm weather.

It is very possible that the use of high-pressure steam in jackets is responsible for many troubles experienced with cutting of cylinders, breaking of rings and increased internal friction. These difficulties can, however, be overcome by proper design and lubrication.

VACUUM.

The maintenance of a good vacuum is a most important aid to economy. This depends on the proper action of air pump and a sufficient supply of circulating water and on the tightness of parts of condenser and exhaust system.

A loss of vacuum may be accounted for by leaks in exhaust pipes whereby air is drawn in. This may be through open exhaust valves of auxiliaries not in use (chief cause), leaks around stuffing boxes of exhaust valves or other valves leading

to condenser, leaky low-pressure piston-rod stuffing boxes; leaks in exhaust joints or joints on condenser, defective air pump. In these cases condenser remains cool.

If vacuum drops and the discharge from air pump is found to be hotter than usual, the circulating water is not sufficient. This may be caused by an extra amount of exhaust steam, a slowing down or stopping of the circulating pump or a stopping up of injection pipes, in which case these pipes will become hot. The division plate in condenser head, or the valves connecting injection and discharge may leak and thus water is allowed to short circuit without passing through condenser tubes. In the above cases the condenser will become hot.

A ready indication of loss of vacuum is the slowing of the engines or a dimness of the electric lights when dynamo exhaust is turned on main condenser. A loss of vacuum shown on gauge without any other indication of change is probably due to the gauge pipe becoming plugged up, or a cock or pipe may be closed.

AUXILIARIES.

The auxiliaries, air, circulating and feed pumps, are likely to give the most trouble, and these require constant attention and oversight.

AIR PUMP.

Separate air pumps, when running very slow, may sometimes stop. Usually the steam end of the air pump gives very little trouble.

The principal trouble with air pumps is the water valves carrying away. If bucket valves are gone the vacuum will drop very materially and the speed of pump will increase. If foot valves go vacuum will not drop but pump will work jerky and hard, vacuum gauge will move with pump. If bead valves are gone vacuum will decrease somewhat and pump will work with a somewhat jerky movement.

Air pumps may become overloaded with water, due to some obstruction in pipes. This will cause them to thump and the vacuum to drop.

The stuffing box of air-pump piston rod should be kept tightly packed with flax packing, and the gland may from time to time require setting up while running.

With independent air pumps, the valve gear must be kept in proper adjustment to give proper length of stroke.

CIRCULATING PUMP.

This, as a rule, only requires regulation and oiling. The speed of the pump should be so regulated that the temperature of the discharge water is a little, not over, 20 degrees below the temperature corresponding to the vacuum that is being maintained. To have this temperature much below the temperature corresponding to the vacuum is simply throwing away heat by means of the circulating water and reducing the temperature of the air-pump discharge.

The temperature corresponding to vacuum is as follows:

28 inches.....100 degrees Fahrenheit.
26 inches.....125 degrees Fahrenheit.
24 inches.....140 degrees Fahrenheit.
22 inches.....152 degrees Fahrenheit.
20 inches.....161 degrees Fehrenheit.

The runner is sometimes likely to come loose on shaft due to key becoming loose. This can be told by a rattling inside the pump chamber. The shafts of pumps are liable to corrosion and become weak and sometimes crack, thus making it necessary to fit new ones.

OPERATION OF FEED AND BILGE PUMPS.

On starting a reciprocating pump, the proper suction and discharge valves should first be opened, making sure that when there are two valves in the connection, both are opened.

Open exhaust and cylinder drains and turn steam on slowly to warm cylinder, then open up gradually. If pump fails to start, move valve gear by hand, observe whether there is an excessive pressure on discharge gauge, if so, a discharge valve may be closed.

If pump starts all right close drains and adjust throttle to suit.

If pump works quickly with a jerky motion the suction may be defective; open pet cock to see whether water is present in suction chamber and ascertain whether there is enough water in tank or bilge to cover end of suction pipe. In some cases it may be necessary to prime pump.

A bilge pump may be primed from the sea, but a feed pump should not be primed from the sea but from some fresh-water tank.

Irregular working may be due to—

Shortness of water in tank.

High temperature of feed suction and accumulation of vapor in pump. To overcome this difficulty the pump may be cooled by putting it on a cold suction for a short time and trying it again, or by cooling it off with a hose.

Sudden change in opening or closing of discharge valves, may be noted by observing the pressure-gauge.

If pump runs fast without an appreciable discharge the trouble may be due to pump valves being carried away or a leaky piston.

Trouble with valve gear.—On every vessel general directions as to the operation of the pumps should be obtained from the manufacturers, for there are many special points peculiar to each type. The drawings of valve gear should be studied so that the operation may be thoroughly understood.

Most difficulties with the valve gear of pumps are caused by neglect and bad treatment. Generally it is the secondary valve which controls the operation of the main steam valve that becomes stuck or fails to move properly. A few general rules as to care of these parts may be given.

a. Remove valve gear frequently and clean off all parts, apply a little kerosene and graphite, the graphite for lubrication and the kerosene to remove rust.

b. Do not keep pump standing for any length of time without moving valve, for it may become rusted in places, due to the presence of moisture.

c. Don't keep drains open when not in operation. Open drains allow access of air and thus allow rusting.

d. Do not supply heavy cylinder oil to valve gear in operation since this oil is liable to cake and cause the valve to stick.

e. Keep pump in continuous operation as much as possible. This applies especially to feed pumps which, if kept going continuously, give little trouble, whereas if intermittently stopped and started are not apt to work well.

The chief trouble experienced is caused by rings of rust or move such obstructions is to take the bonnet off and clean the burrs being formed on the moving parts. The best way to reparts.

In pumps having a valve gear operated by tappet motion and distance nuts the valve stem may be too tightly or too loosely packed so as to necessitate too much force, or, on the other hand, allow the valve to drop of its own weight before it should. (This is especially so in Dow pumps, where a tightening up on valve stem stuffing box will often remedy trouble.) The distance nuts may be not properly set for full length of stroke and the cushioning valve may not be properly adjusted.

Lubrication of pump barrel.—The water cylinders of salt-water pumps require a certain amount of lubrication for both rods and pump barrel. Feed pumps and bilge pumps generally have sufficient oil in the water they handle to give proper lubrication.

Difficulties experienced with the water end of pumps.—Considerable difficulty is found in properly packing the plunger and in securing reliable packing material. Soft packing for this purpose is generally either square braided, flax or square cotton hydraulic. In packing the plunger, especially with cotton packing, care must be taken that the packing is not put in too tight, since it will swell, and this may take place to such an extent that the plunger will stick. In putting on the packing it is advisable to coat it with graphite.

Hard vegetable packing.—This is made of preparations of rubber, lignum vitæ or paper. These are often used and give reasonable satisfaction; they are supplied in the form of rings, and act in a manner similar to piston rings.

Metallic packing for this purpose consists of *brass rings*, which often cause excessive wear, and *antifriction metal rings*, which are more satisfactory, chiefly because they are less likely to wear the surface of the barrel.

Outside-packed plungers are packed with flax packing, and in these pumps special care must be taken to keep the gland free from dirt and grit which would cut the metal of plunger. If plunger becomes cut there is likely to be a considerable loss of fresh water.

Valves of pumps.—Feed pumps use bronze valves with springs and guards. These springs often break and the nuts holding them in place sometimes come loose. In overhauling the pump, special care must be taken to see that these parts are properly secured by split pins and that the springs are set up to the proper tension. The valve seats are in many cases removable and can be taken out and refaced when they become rough. The valves should always have a smooth bearing surface, and in overhauling the tightness of valve should be tested.

Bilge pumps use rubber valves. In time these valves become soft and begin to decompose; this is generally caused by the action of the oil. Rubber valves may be trimmed and turned, but in most cases they should be removed as soon as they become soft or worn.

ECONOMICAL PERFORMANCES.

On the management of engines in connection with regulation of pressure and the adjustment of cut off of H.P. cylinder depends in a great measure the economy during running.

An increase in the number of expansions will increase the economy. Hence for economy the H.P. cut-offs should be run in, but if anywhere near full power is required and cut-offs are in, a limit to the power of the engine is reached, and if the boiler pressure is at the limit the cut-offs will have to be moved out to obtain the increase of power. To get the greatest power with greatest economy the H.P. cut-off should be run in as far as possible, while at the same time using all steam made by boilers. Where the boiler power is considerably in excess

of engine power the cut-offs may be run all the way out and still the boilers will furnish more steam than engine can use. If this is the case the pass-over valves, first the I.P. then the L.P., may be opened and then the limit of power of the engines is reached. There is some question about opening L.P. pass-over since this has the effect of raising the back pressure in L.P. by choking exhaust.

Steam Pressure.—As long as the cut-offs, by being run in, will take care of the steam with throttle open wide, the steam pressure should be kept at the greatest working pressure. The *H.P. cut-off,* and *not* the *throttle,* should regulate the steam.

In working at reduced powers (below ½ power) conditions become such that steam pressure will drop in H.P. cylinder with cut-off all the way in. Then a lowering of pressure should be resorted to until a point is reached at which the H.P. cut-off will take care of steam, maintaining it at a constant pressure for the desired revolutions. The pressure should, however, not be permitted to go below 80 or 90 pounds for triple-expansion engines in order to make sure that they can be properly handled, so that when the pressure is reduced to this with H.P. cut-off run in throttling must be resorted to if it is desired to go slower.

It is thus seen that the working pressure should be kept up and throttle open as long as the H.P. cut-off will take care of steam. When the power is further reduced, the boiler pressure may be reduced, keeping H.P. cut-off in. On reaching a boiler pressure of 90 to 120 pounds, throttling should regulate the supply. In this way the initial pressure will be kept up until cut-off takes place. By keeping a high pressure in steam pipe and throttling, the initial pressure is greatly reduced at point of cut-off and the gain in economy due to the high pressure is lost in a large measure though the steam is superheated by throttling, so that something is here regained.

The smooth working of the engines may govern the position of I.P. and L.P. cut-offs; often when they are run in to limit the engines do not work well.

Running cut-offs in shortens the travel of valve and is likely

to wire draw the steam through ports, and this should not be carried too far.

ADJUSTING LINKS.

Each cylinder should do an equal share of the work. How matters stand in this respect can be told from working up the indicator diagrams, and also roughly by observing the steam pressure in the receivers, the vacuum, and having in mind the cylinder ratios. The drop in pressure in each cylinder multiplied by its ratio to the H.P. should be nearly equal. Taking an engine whose ratio H.P., I.P., L.P. is 1—2, 5—6; pressure in steam pipe, 160; first receiver, 50; second receiver, 10 pounds; vacuum, 20 inches. The drop in H.P. cylinder is 110 pounds, drop in I.P. 40 pounds, drop in L.P. 20 pounds, $40 \times 2\frac{1}{2} = 100$, $20 \times 6 = 120$. In this case, the power of cylinders is somewhere nearly equal, but L.P. is doing the most work. On account of initial condensation an extra allowance should be given the H.P. If in this case the pressures are changed to 48 for first and 7 for second receivers, a more nearly equal division of work would result. When running at low powers, owing to the drop of initial pressures and wire drawing, the drop in the H.P. should be considerably more.

The work done by each cylinder can be regulated by adjusting the link. The following rules for adjusting link may be given.

1. Moving H.P. cut-off makes little or no difference in the distribution of the work.

2. By running cut-off in, that is, cutting off earlier, the work of that cylinder is increased and the work of the cylinder above it decreased, since it produces a higher initial pressure in the receiver of the cylinder on which the cut-off was removed and a higher back pressure on the cylinder above it.

3. By running cut-off out the power of that cylinder is decreased while that of the one above is increased.

4. These effects are produced by altering the pressure in the receivers and not by a change in the ratio of expansion. Moving the I.P. and L.P. cut-offs does not effect the ratio of expansion of the engine as a whole.

From the above it will be observed that altering the cut-off of a cylinder will not change the total amount of work done by it and the cylinder next below, while it does affect the individual power of its cylinder and also the one above it.

To give a few practical examples: Suppose it is found that the L.P. is doing most of the work, the H.P. next and the I.P. least. Move out cut-off of L.P. cylinder. This will decrease the power of that cylinder and increase the power of the I.P. and thus make them both approach to the power of the H.P. If the H. P. is doing most of the work and the I.P. the least, move in the I.P. cut-off. This will decrease the power of H.P. and increase both I.P. and L.P. Now run L.P. cut-off in and the power of L.P. is increased and the I.P. decreased, thus bringing all equal.

The total number of expansions used (neglecting clearance) is found by dividing the decimal of the stroke at which the H.P. cylinder cuts off into the ratio of L.P. to H.P. Thus with H.P. cut-off $= .8$, Ratio $\frac{\text{L.P.}}{\text{H.P.}} = 4$; the number of expansion is 5. If clearance is considered, and to be accurate it should be, the total number of expansions equals

$$\frac{\text{L.P. plus cl}}{\text{K} \times \text{H.P. plus ch}}$$

K equals decimal of cut-off of H.P.
ch equals clearance of H. P.
cl equals clearance of L. P.

Most *four-cylinder triple engines* in U. S. Naval Service are designed so that the sum of power of two L.P. cylinders equals that of either the H.P. or I.P.

RACING.

Racing of engines is caused by rolling and pitching, heeling and the action of the waves on the propeller. Whenever the propeller leaves the water the load upon it is removed and the engine will tend to race. When this happens suddenly there is a danger of breaking something, due to the sudden change

of load on the engines. When racing takes place it is safer to reduce the power of the engines, as then the power is limited and there is not as much danger of engine tearing itself to pieces. When racing becomes excessive, throttling will have to be resorted to. Then the machinist will have to stand by the throttle and, by watching the movement of the ship and the engine, must anticipate when the propellers are lifted out and close the throttle to prevent running away, and, as the propeller dips again, to give more steam.

The best indications are the movements of the engine itself. If it slows down and seems to work hard it shows that the propeller is deep in the water and also that it is likely to be lifted out very soon afterwards, so that while the engine is still working hard is the time to start to close the throttle. After a little practice the engines can be so handled by throttling that no sudden changes in revolutions take place.

WAITING.

If engine is required to be ready at immediate notice everything must be kept in readiness, with auxiliaries running slowly. If there is to be about one-half or an hour's notice steam may be turned off engine.

STOPPING AND WAITING.

For stopping momentarily while maneuvering, the reversing lever is moved to stop and throttle closed, if necessary, with everything ready to move at the next bell. When there is a stop for a longer period, the wicks should be taken out or oil shut off and the air and circulating pumps shut down.

MANAGEMENT OF STEAM.

When stopped or waiting the steam pressure must be looked after. The fireroom must be notified so that fires may be regulated. If pressure rises some auxiliary may be started, such as evaporator, but the bleeder is the direct means of regulating the pressure. The bleeder valves should not be opened suddenly, as the sudden admission of steam is liable to damage

the tubes of the condenser. Special precaution should be taken not to allow safety valves to blow, since there is a loss of fresh water, and generally some difficulty about the valve reseating promptly is encountered.

COMING INTO PORT, SHUTTING DOWN.

When the ship is about to come into port all derangements and maladjustments that have been observed should be examined and their character, position, extent and importance made note of to enable a plan to be made for making repairs. To insure this, each machinist should submit a list of all repairs required in his particular station.

The fires should be allowed to burn down gradually, making sure, however, that there will be sufficient steam for all needs. It is quite important that due notice of arrival in port be obtained so that necessary disposition can be made. All ashes should be hoisted and things prepared for the most important repairs.

When engines are done with fires are hauled or banked as ordered.

The oil and water service should be shut off, all wicks taken out, the drains of engines and steam pipes opened, with bleeder to take care of any surplus steam.

Open the drains to free the machinery of water that is being condensed. Boiler-stop valves should be closed and then bulkhead stops, and every measure taken to drain pipes and engines. To this end jackets and separator should be drained. The air and circulating pumps should not be shut down immediately, but allowed to run for about half an hour in order to cool condenser and draw all moisture out of cylinders. When air and circulating pumps are stopped, all valves and sea connections should be properly closed. The drains should then be closed and internal parts of engine kept free from access of air. When main condenser is shut down fresh water side should be pumped out dry to prevent water going into engines.

In coming into port various dispositions must be made of auxiliary steam and exhaust. If exhaust is on L. P. receiver

it must be turned into condenser before engine is stopped. The auxiliary stop, if fitted, on all boilers not to be kept under steam must be closed. The auxiliary stops of boilers to be kept in use must be open.

Oil cups, pipes and drip pans should be cleaned at once, using hot water and lye to remove oil.

All grease and oil should be cleaned off engines before engines cool and deposit becomes hard. A great deal of dirt can be washed off by using a fresh water hose.

After this crank pits should be cleaned out and bright work attended to. Gaskets and other coverings should be put over ends of bearings and other parts to prevent dirt from dropping in. The stern gland should be tightened up to stop too much water coming in.

The machinery should be put in best position to make the desired repairs, and tools and material got ready for overhauling.

ROUTINE UNDERWAY.

The routine of engine watch underway should be as indicated in a routine table kept on board ship. A sample is here given.

The watches are relieved just before eight bells, it being the general practice to have all the new watch on when eight bells strike. On relieving, all orders should be passed along on each station, and a general statement of how everything is working should be given.

After relieving, a general inspection of everything should be made and defects adjusted.

The machinist in charge of watch should read over the logs of the previous watches to see how matters have been going and whether any new disposition is to be made.

On some ships a condition blank, as per sample, is kept and turned over by the machinist of the watch.

The log data should always be carefully taken, and if it is not possible to be accurately taken, should be left out or noted as such, for misleading or inaccurate data are worse than none.

The hotwell water should be frequently tasted so that it may

U. S. S. MARYLAND

U. S. S. *MARYLAND*.

DAILY AND WEEKLY ROUTINE. ENGINEER DIVISION.

	From "turn to" till twenty minutes to breakfast.	"Turn to" after breakfast (8:15) until noon. Knock off work at 11:30.	After dinner till time for knocking off work of day (this may vary). Until further orders, 4:30.	Evening.
Sunday.....	Finish cleaning stations. 6:00 A. M. "turn to."	Prepare for inspection and inspection by commanding officer. (Preliminary inspection by engineer officer.)	Port watch. (Holiday.)	Evening inspection and reports. Store rooms secure. Water in tanks. D. B. sounded. Lights out. Bilges pumped. W. T. doors secured.
Monday	5:30 to 6:00 A. M. wash clothes. 6:00—7:10 A. M. hoist ashes, move machinery and valves. Detail work and prepare tools. Clean stations.	Muster at stations. Work of day. Drills as per orders.	Work of day.	Ditto.
Tuesday	Ditto.	Ditto.	Ditto.	Ditto.
Wednesday..	Ditto.	Ditto.	Half-holiday. Inspection of clothing, small stores, etc., as directed.	Ditto.
Thursday.....	Ditto.	Ditto.	Work of day.	Ditto.
Friday.....	Clean bilges, move machinery, etc.	Muster at stations. Finished bilges and work of day.	Instructional drills, boats, etc. Inspect and test mechanical devices, compartments, etc.	Ditto.
Saturday.....	Clean stations, move machinery, etc.	Muster at stations. Clean stations; clean out wash rooms for Sunday inspection.	Half-holiday.	Ditto.

MODIFICATIONS.—Section having 12 to 4 A. M. (mid watch) is aux. watch off for day succeeding. This section will have liberty after working hours on liberty days.

Date_____

U. S. S. *MARYLAND.* ENGINEER DEPARTMENT.

CONDITION BLANK.

To be kept in engine room and passed over to relief by engineer of the watch and machinist on watch in port.

Watch							
Boilers banked.....							
Boilers steaming.....							
Coal from bunkers.....							
Reserve feed from.....							
Leak in boilers.....							
Coal allowance.....							
Steam pressure.....							
Revolutions.....							
Evaporators							
Ice machine.....							
Steering engine.....							
Dynamos.....							
Dynamo condenser.....							
Heaters.....							
Pantry circuits.....							
Bath heaters.....							
Auxiliary exhaust on..							
Pumps, feed in use.....							
Pumps, fire and bilge..							
Heated bearings....							

Engine room.

Extra work going on

Fire room.

Officers on duty

_____

Machinist of Watch.

be immediately known when it becomes salty and way by which salt has come in looked for. In addition to tasting the feed water the silver nitrate test may be applied from time to time.

The periodical oiling and feeling of bearings must be kept up, and the round of all the auxiliaries frequently made.

Bilges should be pumped and strainers cleaned during fourth hour, so that the new watch may come on with bilges in good shape.

The double bottoms should be sounded each day at a special time, generally 6 to 8 P. M. watch.

During the time on watch spare time should be used in keeping engine room clean. If this continual cleaning is kept up on the run, there will not be so much work when coming into port, and if an engine room is *never allowed* to get dirty it is more easily kept in good condition.

ACCIDENTS UNDERWAY.

The best prevention for accidents underway is to have everything in proper shape and adjustment so that defects are not likely to develop.

An occasional defect is a great teacher, and the sea experience gained on an old, broken down installation is perhaps the most valuable that could be obtained. When nothing ever breaks, the attendants are not likely to be very familiar with ways of repairing.

Besides the points enumerated under running of engines, the following are among the accidents most likely to occur. With the great improvements in design tending to reliability and freedom from accidents, some of these may very seldom occur.

HOT PISTON ROD.

Hot piston rods are caused by packing being set up too tight, by being out of line, by grit in stuffing box, or insufficient lubrication of the rod.

A hot rod is told by the smell of burning oil and by the fact that the engine appears to work hard, or a thump may develop.

If the rod is quite warm, water or oil applied with a syringe

will hiss and run off in little gloubles (spherical condition), and if very hot, oil will cause a dense smoke.

A hot piston rod is very serious, as there is danger of the rod bending.

If rod is only warm, it may be relieved by slacking back on stuffing-box gland and applying oil liberally and reducing the speed of engines.

If rod is hot the engines should be slowed down to reduce load on piston. The gland must be slackened up to reduce friction. Then oil should be applied with a brush, swabbing all sides of rod so that the different portions may be cooled down at the same time. Water should not be put on a hot rod, as it is liable to cool it suddenly and cause the rod to bend.

With approved kinds of metallic packing, there is not much danger of rods heating. When, however, rods are packed with soft packing and an ordinary gland there may be danger. One of the causes for this is an unequal setting up on packing, whereby one side is too tight and presses hard against the rod.

The packing should not be ragged or too hard, and above all, lubrication must be provided in the packing and also by swabbing rods. There are cases where soft packing (Selden's) has actually caught on fire and caused considerable blaze in the engine room.

One of the principal causes of a piston rod heating is that the piston is thrown out of line with the crosshead slide, in other words, the packing in stuffing box becomes a guide, something it should not be. At top of stroke the entering steam throws the piston against the cylinder and thus throws piston out of line. If the edge of groove in piston becomes burred, the rings will not come to the center again, and thus the piston runs out of line and the rod gets hot.

VALVE RODS.

Valve rods, especially when they pass through guide bushings, sometimes heat and stick, due chiefly to improper lubrication or grit getting on to wearing surface. The remedy is

to cool the parts by a good supply of oil, and, if possible, loosen up on guide.

HOT CONDENSER.

This, of course, will be noticed by a fall in vacuum, and may be caused by a variety of things, the principal thing, a lack of circulating water. When a condenser has become hot due to lack of circulating water, the engine should be stopped and all exhaust turned off the condenser. Otherwise the condenser will get hotter and burn out the packing in glands. The air pump should be continued on the condenser. If the air pump is attached to engine, the feed pump should be put on channel ways to take water out of condenser. If there is any engine-room pump by which water can be put into the condenser and opening the salt drain, a slight circulation can be set up. Usually the water-service pump has some sort of a connection by which this can be done. If there are any connections by which another supply of circulating water can be obtained and condenser thus cooled, the engines may continue to run, but unless this can be done they must be stopped until condenser is again in working order.

The following are some of the different things that may happen to prevent a supply of circulating water.

1. Injection or discharge valves not opened or stopped up. —Sometimes the injection valve may work loose from its stem and remain closed, in which case divers must be sent down to plug opening and then valve may be taken out and repaired. The strainer, if stopped up, will have to be cleared in some way.

2. Runner of circulating pump getting loose from shaft.— This may happen if key gets loose.

3. Shaft of circulating pump breaking due to corrosion.— In these two cases, the pump engines will run without a load.

4. The stopping of pump engine.

LEAKS.

Leakage of steam and fresh water is a matter that should be very carefully looked after; a steam leak not only means the

waste of so much fresh water, but also a loss of heat. Small steam leaks may not appear to have very great importance, but when it is considered that there are hundreds of them and that every particle counts, some idea of what it may amount to can be gathered.

To state roughly, it requires about 4 tons of extra feed per 1,000 horsepower per 24 hours. If more than this is used the cause should be located and leakage stopped.

Leakage is reduced by keeping all joints tight, especially in piping and boilers, by keeping valve spindles properly packed, by keeping valves tight, and by properly packing all stuffing boxes.

AIR PUMP FLOODED.

The air pump may become flooded from a variety of causes. The discharge opening may become obstructed and the water may then not be able to flow away quickly enough. The suction may be suddenly opened and then too much water will come to the pump.

The flooding of the pump will make itself known by heavy thumps and a fall in vacuum. With attached air pump, it would be well to slow down until pump has freed itself of the overload of water, otherwise there is danger of the bonnet of air pump being broken.

With independent air pumps there is less likelihood of pump becoming flooded.

BREAKAGES.

Fractures in cylinders, pistons, piston rods, crossheads and connecting rods are caused by—

1. Water in cylinders.
2. The slacking back of nuts on piston or follower.
3. A dropping down of piston so that the piston strikes at one end.
4. Obstructions such as tools or material left in cylinder.

Aside from fractures caused by priming, the causes above are due to lack of care when engines are overhauled or adjusted, and will thus be seen that it is of extreme importance to see

that all parts, especially nuts, are properly secured and that no obstructions are left in the cylinders. Nuts, bolts, wrenches and the like have sometimes been left in cylinders without causing accident. At other times they have been pressed into the metal of piston and in other cases ground into small bits. The causes and treatment of priming have been discussed under operation of engines.

DISCONNECTING CYLINDERS.

In cases where some of the working parts enumerated above are broken so that they can not be repaired or spares fitted, it may be necessary to disconnect one cylinder and run with the remaining ones.

Where none of the auxiliaries, such as air and circulating pumps are operated from the main engines, any one of the cylinders may be disconnected by taking out or securing its connecting rod, securing piston in place and taking out the valve for that cylinder. In such cases the steam will pass directly from the admission to the exhaust ports and into the next receiver or condenser, as the case may be. In order to prevent loss of steam in the disconnected cylinder by radiation and condensation, the admission ports for that cylinder should be blocked up. Soft wood or melted lead or babbitt may be used for this purpose.

The valve gear in these cases must, of course, be disconnected, the stuffing box for valve stem blanked off and cylinder closed.

ADJUSTING PRESSURES.

A triple-expansion engine can run as a compound with any two cylinders. The cut-off and pressure should, however, be adjusted to suit each cylinder. If the H.P. cylinder is cut out, the boiler pressure should be reduced to the pressure of safety valve on I.P. receiver, or steam must be throttled a suitable amount. If I.P. is cut out a greater range of expansion should be used in the H.P. and the cut-off of L.P. run out so that the pressure on L.P. piston will be the same as before.

PUMPS OPERATED FROM BROKEN CYLINDERS.

When this is the case it is necessary to keep these auxiliaries in operation. If the cylinder and piston are intact and the connecting rod or piston rod broken, these parts may be taken from one of the other cylinders and thus the cylinder kept in operation. If the connecting rods are intact and it is the piston or cylinder that is broken, the cylinder should be disconnected, broken parts removed and piston rod allowed to work through stuffing box as a guide, the pump rods, etc., working from crosshead as before.

ROD BROKEN OR BENT.

If the piston rod is broken or bent it should be disconnected to allow the connecting rod and crosshead to drive the pumps.

RUNNING ENGINE SINGLE CYLINDER.

When it becomes necessary to run an engine with a single cylinder there may be some difficulty at starting to bring her over the center. In order to help matters along, shift valve or eccentric so that you have a big lead on bottom and small cushioning. The weight will then tend to drive her over the top center and the big lead at bottom will give steam to drive her up.

BROKEN CYLINDER HEADS OR CASTINGS.

Where cylinder heads or castings are broken beyond repair they must be disconnected and the ports leading to it blocked up and valve taken out.

BLOCKING PORTS.

For blocking up ports or filling up small holes in a cover or any other part of cylinder where there is no great stress and where the pressure is not great as I.P. and L.P. cylinders, melted lead or babbitt poured into such holes makes a very satisfactory and easily made patch. Where great pressures come upon the blocking, measures will have to be taken to strengthen it sufficiently.

REPAIRING BROKEN HEADS.

Broken heads can usually be repaired in some way.

Heads cracked.—This may generally be repaired by putting a plate on top and securing it by set screws, and, in order to make it steam tight, imbedding it in putty made of red lead, hemp and asbestos or other suitable material. Some sort of plate can generally be found on board, and, if necessary, a door or some other fitting made of plate may be utilized. Patches over broken cylinder covers are often secured by shoring from the deck above.

Head completely broken.—When the head is completely broken up a wooden one may be made, strengthened by plates or iron ribs and placing several courses of sheet packing under it to insure steam tightness. In such cases shoring from above would be advisable. In order to avoid unnecessary shoring, etc., the pressure that is to come upon the cylinder should always be taken into consideration.

BROKEN CYLINDER BOTTOM.

This is, in general, a more difficult job to repair since there is no good opportunity for shoring or securing. In this case it may be advisable to put a patch on the inside by any means available. Then put liners under piston rod to raise it and a distance piece under cylinder cover. To make such repaired joints tight melted babbitt will be very useful.

BROKEN PISTON.

If the piston is not broken into many or small pieces it may be repaired as described by Mr. A. R. Leask (in "Breakdowns at Sea"). In this case a ring of diameter nearly equal to diameter of piston is made and shrunk on the body of piston. The height of this ring should be equal to the difference between the height of piston hub and piston at point where ring is shrunk on. A circular plate is then taken of diameter equal to ring, a hole cut out to fit hub snugly, and ring laid on. Drill several rows of holes into this plate and corresponding

tap holes in piston; then tap in iron rods through holes in plate into piston, cutting off ends and riveting them over plate, or put nuts on top of plate. The repair will then look as sketched in section.

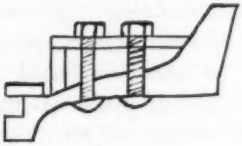


Fig. 1.

The plate should preferably be in one piece, but, if necessary, several may be taken, care being had that the joints do not come over the break in piston.

BROKEN RODS.

Broken rods may be repaired by using the dovetail key (A. R. Leask's "Breakdowns at Sea"). The broken parts are put together and a dovetail (as shown, in sketch) is marked

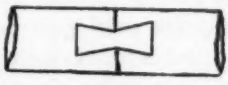


Fig. 2.

thereon, making it about ½ inch larger on one side so as to form a draw or taper for the key. The dovetail is then cut out and cleaned carefully, a corresponding dovetail key made and driven home and riveted over into countersink at smaller end, if necessary, to prevent working back. The key can then be smoothed off and, if necessary, can work through a stuffing box. By this means the rod can be made nearly one-half the original strength.

Fishes and Clamps.—In addition to the dovetail key, where a strong support is desired, fishes and clamps may be used. Here a bar is placed on either side of the broken rod and strong bolted clamps made and bolted up as in sketch. The

number of clamps and length of bar will, of course, always depend upon circumstances. The bars should be placed over the key to prevent it coming out, and on fitting key the direc-

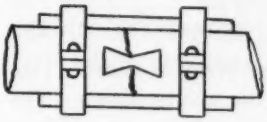


Fig. 3.

tion of the strain brought upon the rod should be considered. In eccentric and connecting rods this strain is athwartship.

BROKEN CRANK PINS.

A break in the shaft may occur at the fillets or at fracture shown in sketch. The best way to repair such a fracture would be to bore a hole through crank pin and webs and insert a pin to fit tightly into the hole. The size of such pin should be about three-fourths the diameter of the crank pin, provided the area of pin does not exceed one-third area of web.

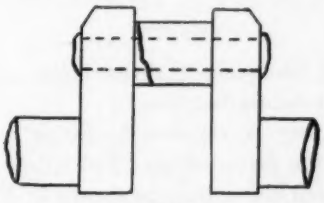


Fig. 4.

To bore the hole, first use an ordinary drill, then rig a boring bar and take successive cuts; as the hole increases use a larger boring bar and stronger tools. A wooden bush to prevent bar getting out of line will be useful. The pin, being prepared to fit the hole closely, should be driven in hard and the ends riveted over.

If anything like full power is desired, the repaired crank should be shifted to the forward cylinder, since the after crank has to stand the twisting movement of the whole engine while the forward has only that due to one cylinder.

The original surface of crank pin must be retained to give the bearing surface, in case this is broken out or rough it may be smoothed up with white metal and then scraped.

BROKEN CRANK WEB.

Crank shafts may break at webs. The repair for this it to put a strap around the web as shown. The size of strap should

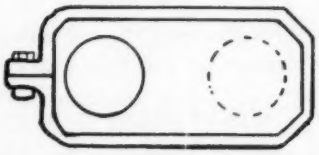


Fig. 5.

be such that the area of section of strap is about one-quarter area section of web. The strap should be forged to shape, the holes bored for bolt, and then should be put on hot, tightened up and allowed to cool. The strap will then firmly grip the web.

SHAFTS.

A fracture may take place at thrust collar, and an example of a repair of this nature is given.

Generally this may be repaired by fitting a clamp around the shaft to keep the parts together and fitting bolts with distance pieces between the collars as shown in Fig. 6.

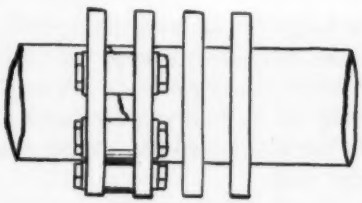


Fig. 6.

Bolts or some other means of bringing the two parts together tightly must be provided.

When the break extends over several collars, long bolts must be used. It is also better to use a larger number of small bolts than a few large ones.

A break in the line shafting is harder to repair. Means must be provided, first for bringing the two parts together, and then material for taking the twisting movement. A

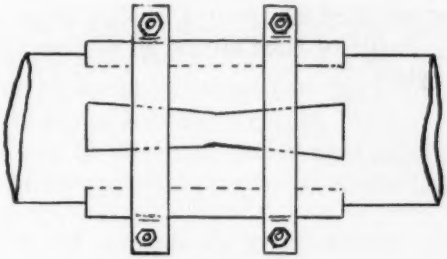


Fig. 7.

method of making this repair is given on page 130, Leask, "Breakdowns at Sea." See Fig. 7.

The dovetail keys keep the parts of shaft together and the

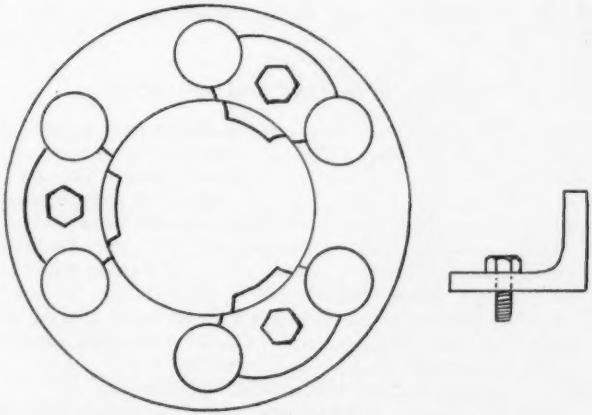


Fig. 8.

clamps bind the keys close to the shaft. To fit clamp so that they come up real tight, the bolts should be put on hot and nuts

set up; as they cool the bolts will contract and grip the shaft securely.

A shaft broken at a coupling is a more serious repair. Such a repair may be accomplished as shown by sketch taken from pages 127 and 128, Leask, "Breakdowns at Sea," Fig. 8. The angle plates are cut out so as to fit exactly between the bolt heads and the other edges into a recess cut into shaft. The angle plates are fitted and secured in place by set screws. A stout clamp should be fitted around all of them to keep them securely in place.

CONTRACT TRIALS OF U. S. BATTLESHIP *VIRGINIA.*

The *Virginia* is one of the three battleships authorized by Act of Congress approved March 3, 1899, and was built by the Newport News Shipbuilding and Dry Dock Company under contract signed February 15, 1901.

The total cost of the vessel, excluding armament, certain portions of the armor, etc., is $3,590,000, of which $1,240,000 is for machinery.

The guaranteed speed is 19 knots, maintained for four consecutive hours, on a mean draught of 23 feet 9 inches, corresponding to a displacement of about 15,000 tons.

As a detailed description of the *Virginia* appeared in the last issue of the JOURNAL (pp. 1,113 to 1,132), the principal dimensions only will be given here.

The principal dimensions and characteristics of the hull are as follows:

PRINCIPAL DIMENSIONS OF HULL.

Length between perpendiculars, feet	435
on load-water line, feet	435
over all, feet and inches	441- 3
Beam, extreme, feet and inches	76- 2½
at load-water line, feet and inches	76- 2½
molded, feet	76
Ratio of length to beam	5.708
Depth, molded, main-deck side at M.S., feet and inches	41- 3⅜
Draught, normal, mean, feet and inches	23- 9
Displacement, mean load, tons	14,979
full load tons	16.094
per inch at L.W.L., tons	61.1
Area of midship section to L.W.L., square feet	1,725
L.W.L. plane, square feet	23,650
Wetted surface, square feet	39,200
Center of gravity of L.W.L. plane aft of M.S., feet and inches	3- 3½
above bottom of keel, feet	26.25

Center of buoyancy above bottom of keel, feet	13
forward of midships, feet	105
Transverse metacenter above center of buoyancy, feet	19.18
Longitudinal metacenter above center of buoyancy, feet	519
Number of frames	108
Coefficient of fineness, block	.6696
M.S.	.9618
L.W.L.	.7724

ARMAMENT.

Four 12-inch guns are mounted in pairs in two elliptical, electrically-controlled, balanced turrets, each pair of guns having an arc of fire of 270 degrees.

Eight 8-inch guns are mounted in pairs in four elliptical turrets, two of which are superposed on the 12-inch turrets, forming part thereof.

The other two 8-inch turrets are balanced, electrically controlled and placed one on each beam; each pair of guns so mounted having an arc of fire of 180 degrees.

There are twelve 6-inch R.F. guns mounted singly on pedestal mounts, with shields, in a central armored casemate on the gun deck, each gun having an arc of fire of 110 degrees.

The secondary battery consists of twelve 3-inch R.F. guns, twelve 3-pounder semi-automatic guns, four 1-pounder heavy automatic guns, four 1-pounder heavy R.F. guns, two 3-inch field guns, two .30-caliber machine guns, and six .30-caliber automatic guns.

There are four submerged torpedo tubes, two forward and two aft.

PROPELLING MACHINERY.

There are two four-cylinder, triple-expansion, inward-turning engines of the vertical, inverted, direct-acting type in two watertight compartments.

Steam is generated by twenty-four Niclausse boilers, with a total grate surface of 1,459 square feet, and heating surface of 57,534 square feet.

The propellers are three-bladed, manganese-bronze, inboard turning, the starboard one being left-handed. They are

designed as true screws, and the surfaces of the blades are polished and painted.

Number of blades	3
Diameter, feet and inches	17- 3
of hub, feet and inches	4- 6
Length of hub, feet and inches	3- 6
Helicoidal area, square feet	84.5
Disc area, square feet	233.71
Ratio of pitch to diameter	1.058
Pitch (adjustable one foot either way), feet and inches	18- 3

STANDARDIZATION TRIALS.

For the purpose of standardizing the screws runs were made over the measured-mile course off Rockland, Maine, November 21, 1905. The weather was clear and cold, with moderate to fresh breeze, and smooth sea.

Previous to the vessel's starting on the standardization runs the draught was taken, and found to be—

Forward, feet and inches, ...	23- 8
Aft, feet and inches, ...	23-11
Corresponding displacement, tons, . . .	15,010

These runs gave the data recorded in Tables I and II, from which the curves in Plate I were drawn.

After the standardization runs the draught was found to be—

Forward, feet and inches, ...	23-03
Aft, feet and inches, ...	23-09½
Corresponding displacement, tons, . . .	14,805

OFFICIAL FOUR-HOURS' TRIAL.

On November 23, 1905, the *Virginia* got under way at 7·30 A. M. for the official four-hours' trial. The weather was overcast and cool with gentle to moderate breezes. The sea was smooth to moderate. At 9·13 A. M. the trial commenced, finishing at 1·13 P. M., completing the four hours required by the contract.

The average revolutions per minute during the trial were 129.57, corresponding to a speed of 19.01 knots.

The draught and displacement at the beginning of the trial were—

Forward, feet and inches,	23–08¾
Aft, feet and inches,	24–00¼
Mean, feet and inches,	23–10½
Corresponding displacement,	15,070

It was estimated that the draught at the middle of the run was 23 feet 9 inches.

A synopsis of the data obtained follows.

PERFORMANCE.—FOUR-HOURS' OFFICIAL TRIAL.

Steam pressures (*Average one-half hourly observations*).

	Starboard.	Port.
Steam pressure at boilers (per gauge), pounds	260	
H.P. steam (per gauge), pounds	227.4	229.6
1st receiver (absolute), pounds	114.2	122
2d receiver (absolute), pounds	38.8	39.6
Vacuum in condensers, inches of mercury	27	27.2

Mean effective pressures in cylinders, in pounds per square inch.

Main engine:

Starboard.....H.P., 97.3; I.P., 54.1; F.L.P., 17.7; A.L.P., 19.8

Port.....H.P., 82.3; I.P., 58.46; F.L.P., 20.06; A.L.P., 20.01

Mean pressure, in pounds per square inch on L.P. piston, equivalent to aggregate M.E.P. on all pistons, starboard..... 52.32

Mean pressure, in pounds per square inch on L.P. piston, equivalent to aggregate M.E.P. on all pistons, port..... 53.14

Revolutions per minute.

Main engines.....starboard, 129.6; port, 129.54

Pumps, air.....starboard, 19.8; port, 22.3

circulating.....starboard, 169.7; port, 167

feed, double strokes per minute.....starboard, inboard, outboard, 26.6

port, inboard, outboard, 35.7

Blower engines, average of twelve blowers..... 528

Speed of ship, in knots per hour..... 19.01

Slip of propeller, in per cent. of its own speed, based on mean pitch, starboard, 20.07; port, 20.04

Air pressure in firerooms, in inches of water..... 1.78

Indicated Horsepower.

Main engines :		
Starboard..H.P., 2,863.8 ; I.P., 4,296.3 ; F.L.P., 1,887.2 ; A.L.P.,		2,117.9
Total		11,165.2
Port.....H.P., 2,421.5 ; I.P., 4,640.7 ; F.L.P., 2,139.6 ; A.L.P.,		2,134.2
Total		11,336
Grand total		22,501.2
Air pump	starboard, 11.75 ; port,	13.3
Circulating pump	starboard, 56 ; port,	53
Feed pumps	starboard, 69.5 ; port,	102
Hotwell pumps	starboard, 18.5 ; port,	15.9
Fire and bilge pump	starboard, 7.8 ; port,	4.7
Blower engines, for forced draft in firerooms, 12 blowers		264
Dynamo engines		80
Collective of all main engines		22,501.2
main engines, air, circulating and feed pumps		22,841.15
and auxiliary engines in operation on trial		23,468.45
all machinery during trial, per square foot of G.S.		16.078
H.S.		.408

COAL.

Kind and quality	New River.
Pounds per hour	68,507
Pounds per hour per I.H.P., collective, all machinery in operation	2.9
Pounds per hour per I.H.P., collective, main engines, air, circulating and feed pumps	2.999
Pounds per hour per square foot of G.S.	46.932
Pounds per hour per square foot of H.S.	1.191
Cooling surface, square feet per I.H.P.	1.020
Heating surface, square feet per I.H.P.	2.452

The working of the machinery, both main and auxiliary, and its performance during the trial, were entirely satisfactory, except the boilers, from which there was an excessive radiation of heat, both from casings and uptakes. The contractors expect to overcome this objectionable feature by the installation of additional covering and a modification in the baffling.

Table I.—U. S. S. *VIRGINIA.*

Data from Progressive Runs Over Measured-Mile Course off Rockland, Maine, November 21, 1905.

Number and direction of run.	Time on course.	Revolutions of main engines.					Speed in knots per hour.	Indicated horsepower.											
		To make 1 knot.		Per minute.				Starboard engine.					Port engine.				Total of both main engines.	Auxiliaries in operation	Total all machinery.
		Starboard.	Port.	Starboard.	Port.	Mean.		H.P. cylinder.	I.P. cylinder.	F.L.P. cylinder.	A.L.P. cylinder.	Total.	H.P. cylinder.	I.P. cylinder.	F.L.P. cylinder.	A.L.P. cylinder.	Total.			
	m. s.																		
1-S.	3-2.5	390.6	389.45	128.45	128.07	128.26	19.731	3,200	3,658.5	1,681.5	1,903.5	10,443.5	3,112.0	3,844.0	1,708.0	1,856.0	10,520.0	20,963.5	833.75	21,797.3
2-N.	3-18.4	424.3	424.95	128.38	128.57	128.47	18.152	3,527.8	3,695.8	1,701.7	2,008.2	10,933.5	3,109.6	4,055.6	1,905.3	1,905.3	10,975.8	21,909.3	818.28	22,727.6
3-S.	3-2.4	393.55	394.15	129.49	129.69	129.59	19.742	3,234.9	3,807.0	1,759.0	1,961.6	10,762 5	2,709.7	4,527.8	1,943.3	1,868.5	11,049.3	21,811.8	824.88	22,636.7
4-N.	3-16.6	426.20	420.05	130.07	128.19	129.13	18.311	2,998.3	4,501.3	1,927.5	2,141.8	11,568.9	2,954.9	4,082 9	1,910.3	1,846.9	10,795.0	22,363.9	781.33	23,145 2
5-S.	3-2.8	398.40	394.75	130.91	129.71	130.31	19.694	2,958.0	4,530.3	1,961.6	2,177.1	11,627.0	2,930.9	4,250.4	1,879.5	1,986.3	11,047.1	22,674.1	795.26	23,469.4
6-N.	3-18.4	414 35	414.45	125.18	125.21	125.20	18.145	3,596.2	3,105.3	1,308.8	1,453.1	9,463.4	3,412·2	3,489.4	1,391.6	1,494.8	9,788.0	19,251.4	660.03	19,911.4
7-S.	3-11.9	383.55	381.15	119.96	119.20	119.58	18 760	3,242.0	2,814.2	1,303.7	1,560.5	8,920.4	2,977.8	2,847.4	1,099 1	1,305.2	8,229.5	17,149.9	483 20	17,633.1
8-N.	3-27.8	396.90	397.35	114.63	114.76	114.70	17 329	2,844.0	2,281.8	1,038 1	1,283.5	7,447.4	2,710.4	2,530.4	1,218.8	1,152.7	7,612.3	15,059.7	455.00	15,514.7
9-S.	3-21 6	376 95	376.80	112.30	112.25	112.28	17.857	2,346.4	2,132.3	850.6	1,174.2	6,503.5	2,510.9	2,275.7	1,136 7	998.1	6,921.4	13,424 9	424.70	13,849.6
10-N.	3-38.1	398 45	394.55	109.62	108.54	109.08	16.506	2,452.2	1,953.8	794.2	1,065.0	6,265.2	2,317.0	1,941.2	875.7	884.7	6,018.6	12,283.8	385.30	12,669 1
11-S.	3-31.0	377.15	375.85	107.30	106.93	107.12	17.062	2,266.2	1,761.3	795.0	892.2	5,714.7	2,161.2	1,899.3	792.3	827.5	5,680.3	11,395.0	390.00	11,785.0
12-N.	3-47.0	386.00	388.45	102.05	102.70	102.38	15.863	1,947.7	2,187.7	697.4	756.1	5,588.9	1,959.2	1,704.7	701.8	710.2	5,075.9	10,664.8	404.73	11,069.5

Table II.—U. S. S. *VIRGINIA.*

Data from Progressive Runs over Measured-Mile Course off Rockland, Maine, November 21, 1905.

Number and direction of run.	Which engine.	Steam pressure.			Mean effective pressure.				Vacuum, in inches.	Air pressure, in inches of water.	Cut-off in cylinders from beginning.			Temperatures.						Revolutions or double strokes.						
		In boilers above atmosphere.	At engine above atmosphere.	H. P. steam chest, absolute.	In 1st receiver, absolute.	In 2d receiver, absolute.	H. P. cylinder.	I. P. cylinder.	Forward L. P. cylinder.	After L. P. cylinder.	Referred to L. P.		H. P.	I. P.	Forward L. P.	After L. P.	Outside air.	Engine-room working platform.	Firerom working level	Injection.	Discharge.	Hotwell.	Feed.	Main feed pump.	Hotwell pump.	Sanitary pump.	Bilge pump.	Air pump.	Firerom blowers.	Circulating pump.	Pyrometer, height 46 feet above grates.
1–S.	S.	262	250	255	105	35	109.7	46.5	15.9	18	49.38	27	.92	.77	.77	.77	.77	38	77	70	48	116	86	159	17	33	53	12	477.5	176	770
	P.	262	245	255	100	34	107	49	16.2	17.6	49 89	27.2		.77	.77	.77	.77		78	70	52	127	80		38	30	72	10		147	
2–N.	S.	268	255	260	105	35	121	47	16.1	19	51.72	27.3	1.21	.77	.77	.77	.77	40	75	70	49	115	97	167	38.5	30	58	15	479	160	893
	P.	268	250	260	102	36.5	106.5	51.5	18	18.4	50.29	27		.77	.77	.77	.77		80	70	51	132	84		38	22	76	11		156	
3–S.	S.	263	245	255	105	36	110	48	16.5	18.4	50.47	27.2	.93	.77	.77	.77	.77	46	76	71	48	113	97	160	32.5	37	58	15	475	170	946
	P.	263	240	250	119	37	92	37	18.2	17.5	51.74	27		.77	.77	.77	.77		78	71	51	130	92		38	28	88	10		154	
4–N.	S.	263	245	255	120	40	101.5	56.5	18	20	54.02	27.3	1.27	.77	.77	.77	.77	46	78	70	49	116	95	164	19	38	58	15	463	176	930
	P.	263	243	255	103	36.8	101.5	58	18.1	17.5	51.13	28.6		.77	.77	.77	.77		77	70	51	136	88		37	26	76	15		152	
5–S.	S.	264	250	255	115	38	99.5	56.5	18.2	20.2	53.94	27.4	1.11	.77	.77	.77	.77	48	77	72	49	111	85	162	21	38	63	15	464	184	1053
	P.	264	245	255	117	36	99.5	53.5	17.6	18.6	51.72	27.2		.77	.77	.77	.77		80	72	51	127	102		37.5	26	63	15		156	
6–N.	S.	266	250	260	90	32	126.5	46.5	12.7	14.1	45.92	26 1	1 11	.77	.77	.77	.77	46	77	71	48	120	96	165	14	36	49	15	412	180	1037
	P.	266	245	255	89	31	120	45 5	13.5	14.5	47 47	27.4		.77	.77	.77	.77		80	71	51	130	92		38	15	64	14		164	
7–S.	S.	259	245	255	81	28	119	38.3	13.2	15.8	45.16	28.3	.47	.77	.77	.77	.77	46	76	75	48	116	84	150	7.5	30	61	16	333	168	965
	P.	259	245	241	84	26	110	39	11.2	13.3	41.93	27.5		.77	.77	.77	.77		78	75	52	128	88		31	20	54	17		154	
8–N.	S.	260	247	255	79	24	109	32.5	11	13	39.42	28.5	.49	.77	.77	.77	.77	46	76	75	47	98	88	150	9	16	53	16	288	170	967
	P.	260	247	255	78	24	104	36	12.9	12.2	40.28	28		.77	.77	.77	.77		76	75	51	115	82		6	16	53	16		156	
9–S.	S.	257	250	255	65	20	92	31	9.2	12 7	35.17	28	.18	.77	.77	.77	.77	50	76	77	48	108	88	150	30	18	59	13	257	154	865
	P.	257	252	190	70	21	98.5	33.1	12.3	10.8	37.94	28.5		.77	.77	.77	.77		78	77	51	100	80		35	9	65	18		150	
10–N.	S.	244	225	190	62	19	98.5	29.1	9.8	11.8	34.71	27.2	.19	.77	.77	.77	.77	50	76	78	48	116	93	160	24	43	38	12	248	140	702
	P.	244	226	175	64	19.5	94	29.2	9.8	9.9	33.67	28.3		.77	.77	.77	.77		78	78	51	103	70		18	20	36	14		148	
11–S.	S.	253	240	180	57	20	93	26.8	9	10.1	32.35	28	.15	.77	.77	.77	.77	50	76	77	48	92	84	110	12.5	28	52	14	270	170	623
	P.	253	242	168	61	19.2	89	29	9	9.4	32.25	27.7		.77	.77	.77	.77		79	77	52	108	66		14	15	66	16		150	
12–N.	S.	248	238	165	53	18.5	86	35	8.3	9	33.53	28	.32	.77	.77	.77	.77	52	75	77	48	94	76	100	21	19	43	15	274	160	577
	P.	248	237	162	58	17.2	84	27.1	8.3	8.4	30.01	27.6		.77	.77	.77	.77		79	77	52	102	64		8	21	92	17		146	

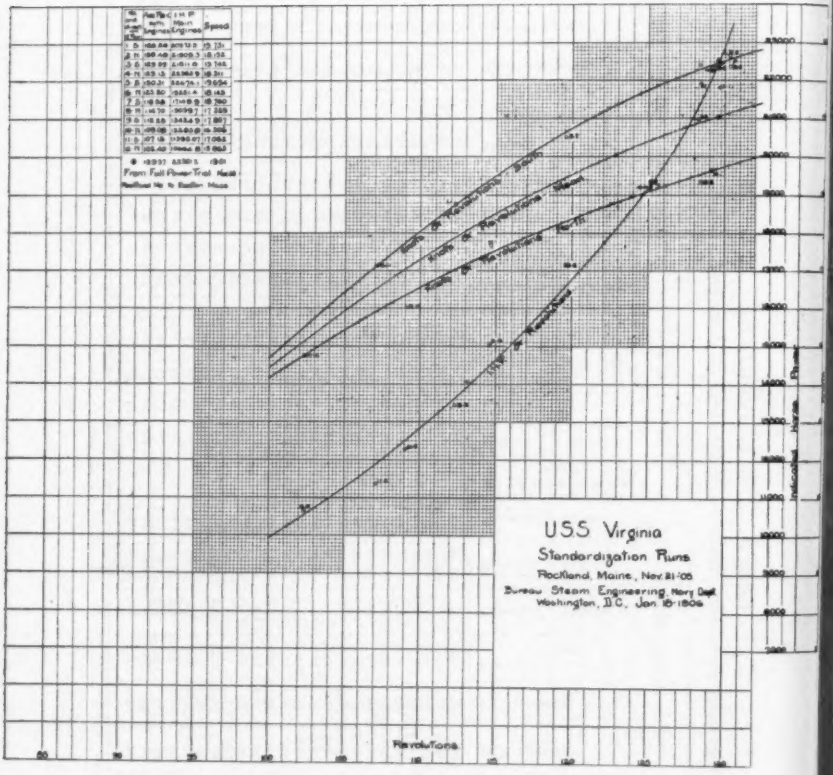
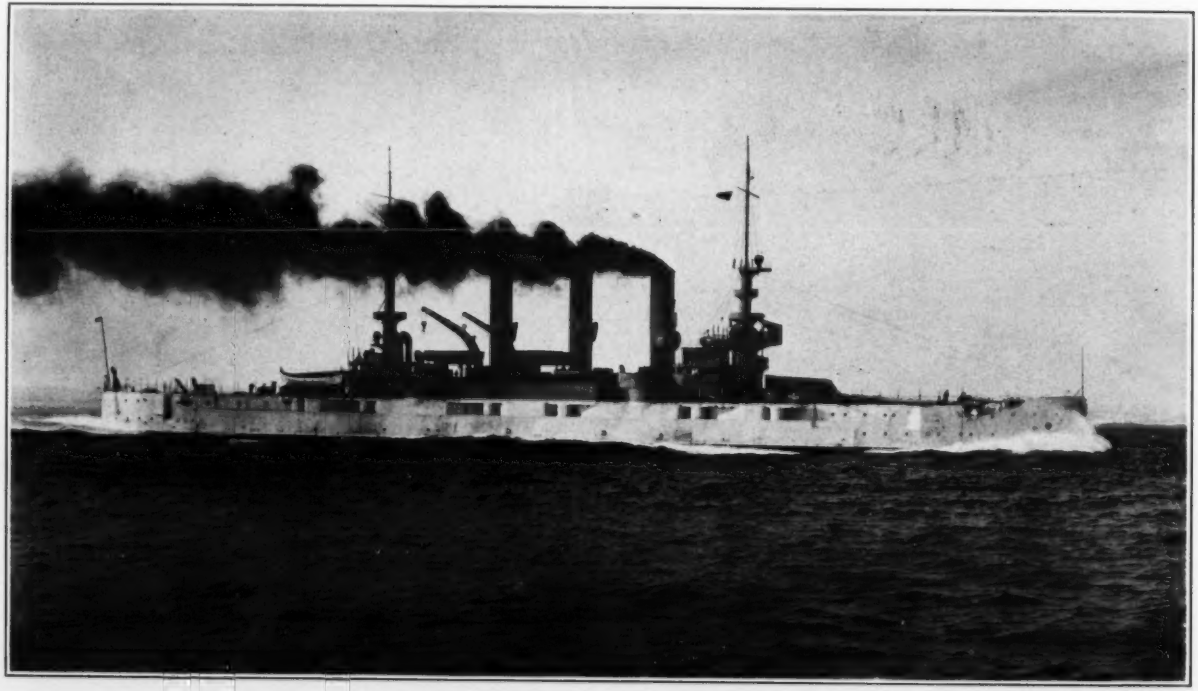
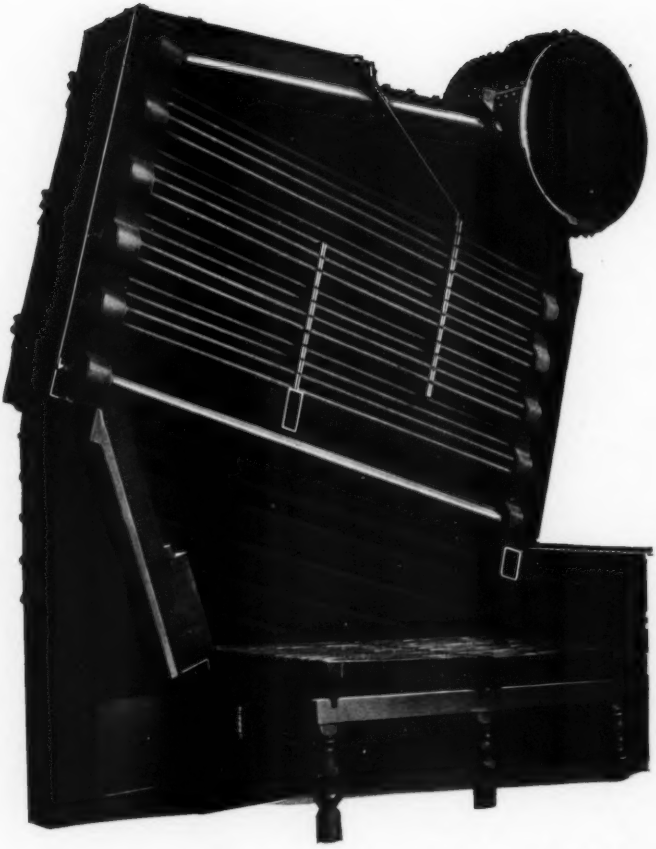

USS Virginia
Standardization Runs
Rockland, Maine, Nov. 21-'05
Bureau Steam Engineering, Navy Dept.
Washington, D.C., Jan. 16-1906
Revolutions
Indicated Horse Power

Plate I.



U. S. Battleship "Louisiana."



SECTION OF BABCOCK & WILCOX BOILER OF "LOUISIANA."

THE U. S. BATTLESHIP *LOUISIANA*.

DESCRIPTION AND TRIALS.

By Lieutenant R. K. Crank, U. S. Navy, Member.

The *Louisiana* is one of two battleships authorized by Act of Congress approved July 1, 1902; the other, the sister ship, being the *Connecticut*, building at the Navy Yard, Brooklyn, N. Y.

The under-water body of each of these two vessels was designed in accordance with the results of careful experiments made with models in the model tanks at the Washington Navy Yard, and these ships are the first large vessels of war of the U. S. Navy in the design of which this method was followed. The ease with which the *Louisiana* met the contract requirements for speed, the performance of the ship, in exceeding by 0.823 knot her contract speed; the close agreement between the horsepower and assumed speeds, as found by the tank trials, and the horsepower and speed on actual trial of the ship, all fully justify the building of the tank, if such justification were ever needed. To judge from the expressions of approval and the praise and enthusiasm of those who were present at the trial, the *Louisiana* is the best battleship that has been turned out in the United States.

The *Louisiana* is building at the yard of the Newport News Shipbuilding and Dry Dock Company, under a contract signed October 15, 1902, which provides that the ship shall be delivered within forty-one months, or on March 15, 1906; up to January 1, 1906, no extension of time had been granted the contractors.

The total contract cost of the vessel, excluding armament, is $3,990,000, of which $1,000,000 is for propelling machinery and other items under Steam Engineering.

The guaranteed speed is eighteen knots per hour, to be maintained during four consecutive hours, under forced draft, the air pressure not to exceed two inches of water; the vessel to be on a mean draught of 24 feet 6 inches, the corresponding displacement being 16,000 tons. The contract imposes penalties for failure to come up to speed as follows: a deduction at the rate of $50,000 for one quarter knot below 18; at the rate of $100,000 for one quarter knot below 17¾ and, if the speed falls below 17½, the Government to have the right to reject the vessel.

The actual performance of the ship on her official four-hour run, on December 13, 1905, at Rockland, Maine, was 18.823 knots, at a mean draught of 24 feet 7¼ inches, corresponding to a displacement of 16,055 tons, with a mean air pressure of 1.98 inches.

The keel was laid on February 7, 1903, and the vessel was launched on August 27, 1904.

COMPLEMENT.

Admiral	1	Seaman branch	401
Chief of staff	1	Artificer branch	45
Commanding officer	1	Engine-room branch	226
Wardroom officers	19	Special branch	33
Junior officers	20	Messmen branch	27
Warrant officers	9	Commissary branch	17
	—	Marines	72
	51	Additional for flag	19
	840		—
	—		840
Total	891		

GENERAL HULL DATA.*

Length between perpendiculars, feet and inches	450- 0
Projection forward of F.P., feet and inches	6- 4
aft of A.P., feet and inches	0- 0
Length over all, feet and inches	456- 4

*The writer is indebted to Naval Constructor Henry Williams, U. S. N., for the data and information regarding the hull.

Length on L.W.L., feet and inches	450– 0
of straight keel, feet and inches	358– 0
Breadth, molded, feet and inches	76– 5½
extreme, feet and inches	76– 10
Depth, molded, main deck side at M.S. (56½), feet and inches	43– 1⅝
Draught, feet and inches	24– 6
Ratio length to beam	5.856
Displacement, normal, 24 feet 6 inches draught, tons	15,970
(est. 900 tons coal, 50 tons reserve feed water).	
Area midship section, square feet	1,808
L.W.L plane, square feet	26,560
wetted surface, square feet	44,500
C.G. L.W.L. plane aft of M.S. (56½), feet	4.4
C.B. above bottom of keel, feet and inches	13– 4
aft of M.S. (56½), feet	.765
Tranverse metacenter above C.B., feet and inches	18– 9½
Longitudinal metacenter above C.B., feet and inches	538– 6
Coefficient of fineness, block	.6592
mid section	.9604
L.W.L. plane	.7683
cylindrical	.6873

Shafts incline up and aft at angle of 55 minutes 42 seconds.
diverge at angle with center line of 35 minutes 48.7 seconds.
center line at 72 { 10 minutes 6 seconds from center line of ship. / 7 minutes ⅝ second from base line.
108 { 12 minutes from center line of ship. / 9 minutes 4⅝ seconds from base line.
rake up .19446 seconds per foot; out .125 seconds per foot.
Center line meet.

Screws, diameter, feet and inches	17– 3
pitch, adjustable 17 feet to 19 feet, feet and inches	18– 0
area expanded, square feet	84.5
projected, square feet	71.5
center forward of A.P. (26 inches aft 108) feet and inches	16– 10
Rudder axis forward of A.P. (3 feet 9 inches aft 108) feet and inches	15– 3
Strut center line forward of A.P. (20 inches forward 108), feet and inches	20– 8
No. of frames (spaced 4 feet throughout)	112
watertight compartments	310

GENERAL DESCRIPTION.

The hull is made of basic, open-hearth steel, with frames spaced 4 feet apart, except under the engine-compartment, where the frames are spaced 2 feet apart.

The inner bottom extends from frame 12 to frame 98.

The double-bottom compartments between frames 58½ and 67½ are arranged for use as reserve feed tanks. A coffer-dam, 30 inches in width, extends the entire length of the ship about the protective deck, being carried to a height of 3 feet above the protective deck, and is packed with corn-pith cellulose in those parts forward and abaft the midship armor. All unexposed decks throughout the ship are covered with linoleum, except in the magazines, in certain store rooms, wash rooms, water closets, etc., where a wood covering placed on tiling is used.

The Superstructure.—The superstructure, built on the main deck, extends from frame 35 to a point 21 inches abaft frame 79, or between the 12-inch turrets; it extends the full width of the ship between the forward and after pairs of 8-inch turrets; it is recessed to make room for these turrets, and is cut away to permit a dead-ahead fire for the forward pair and a dead-astern fire for the after pair of 8-inch turrets. It is also cut away at the corners to give an arc of fire of 135 degrees for the 12-inch guns, forward and aft.

The Bridge Deck.—The bridge deck forms the top of the superstructure. Over the forward end is built the forward bridge, on which is the chart house, emergency cabin, entrance to conning-tower and the torpedo-directing station, which is joined to the forward side of the conning tower. Over the forward bridge is the usual flying bridge, on which are the steering compass, two search lights and the usual navigating instruments. The standard compass is mounted on top of the chart house.

On the after end of the bridge deck is the after bridge, from which is the entrance to the armored signal tower.

Top Hamper on the Superstructure.—There is one electric boat crane on each side of the ship, of 15-tons capacity, located abreast of the after funnel. The boats, with the exception of the 30-foot whaleboats, 30-foot barge, 30-foot gig whaleboat, and two 30-foot cutters, which are on the main deck davits, are stowed on the bridge deck. The dinghies are located abaft the after funnel, between the recesses for the

after pair of 8-inch turrets, the 16-foot dinghy nested in a 20-foot dinghy on port side near center of ship, and the 14-foot dinghy in the other 20-foot dinghy on starboard side near center line. The other boats are stowed abreast in wake of the middle and after funnels. Beginning outboard, those located under starboard crane are, one 50-foot steam cutter, which is temporarily replaced by a 40-foot steam cutter; one 36-foot steam cutter, and one 30-foot cutter which is nested in a 33-foot launch. Under the port crane are, one 36-foot steam cutter, a nest of one 30-foot cutter, one 33-foot launch and one 36-foot launch, and a nest of one 30-foot cutter and a 33-foot launch.

Main Deck.—The main deck extends the full length of the vessel, the superstructure partially covering it, as described above. At the bow is a waist plate, 3 feet 6 inches high, extending aft to frame 7.

Anchors, etc.—There is one anchor crane on each side, for handling Navy-type sheet anchors, bill boards for which are located between frames 9 and 13. There is, however, but one Navy-type anchor provided, which weighs 17,490 pounds, and which is stowed on the starboard bill board. The tackles for these cranes are operated by electric winches abreast windlass house. There are two hawsepipes on each side, opening through this deck onto the forecastle, from which the chains have a direct lead to the windlass house, which is in front of the forward 12-inch turret and projects above the deck so that the chains enter the front, directly to the wildcat. In the forward hawsepipes are carried two 17,350-pound Baldt stockless anchors. There are also two kedge anchors, weighing 810 and 410 pounds, carried forward, and a 5,655-pound stern anchor and a 4,160-pound stream anchor, carried aft, all stowed against plating of superstructure. Forward, two 20-foot dinghies are stowed for harbor position only and two 30-foot cutters permanently, and aft there are two 30-foot whale boats, a 30-foot barge and a 30-foot gig whaleboat.

Within the superstructure, and at its forward end, is the crew's galley and mess issuing room; amidships is the officers'

galley, and aft are the general mess pantry, paymaster's and executive officer's offices, batallion lockers and wireless telegraph room. Drying rooms are located around all funnels.

GUN DECK.

Forward, on this deck, are located the crew's showers and washroom, petty officers' and machinists' water closets, and the staterooms of the sergeant of marines and master-at-arms. Between the 12-inch barbettes is the 7-inch battery, in which space are located the bakery, printing office, brigs, junior, warrant and wardroom officers' lavatories, baths and shower-baths; armory and engineer's, navigator's and ordnance officer's offices. Abaft of the after 12-inch barbette are the admiral's, captain's and chief-of-staff's quarters and seven wardroom staterooms.

The admiral's quarters are unusually spacious, and, with the quarters of the chief of staff, take up nearly the entire after part of this deck. With the increasing numbers of the fleet-staff, it would be impossible to find quarters for the necessary officers were the *Louisiana* to become a flagship.

Due to the installation of 7-inch guns on this deck, the deck height is greater than has been previously customary, being 8 feet 7 inches between decks and 7 feet 9 inches to bottom of beams. This necessitated special arrangements for the hammock hooks, which extend about 7 inches below bottom of beams.

BERTH DECK.

On this deck are located, beginning forward, lamp rooms, stowage for oils, paint-mixing rooms, commissary stores and band rooms, the isolation ward, sick bay and its bath, dispensary and operating room. The machinists' and petty officers' quarters and washrooms and the refrigerating plant are abreast the forward 12-inch barbette; adjoining these are the steam laundry, general mess stores and paymaster's issuing room. The firemen's and servants' washrooms are on the center line between uptakes and take up their entire length. Abaft of uptake is the evaporator room. The general work-

shop is over the engine space, between hatches. Abreast of the engine-room hatches and after barbette are the junior and warrant officers' quarters and mess rooms, on starboard and port sides respectively. Abaft, are the wardroom officers' quarters, and away aft several showers are provided.

PROTECTIVE DECK.

This deck extends the full length of the ship and is 3 inches thick on the slopes and 1½ inches on the flat. The cofferdam previously mentioned extends its entire length, and the spaces between the slopes and berth deck are used for stores, except from frame 40½ to frame 72, where the berth-deck coal bunkers extend to this deck.

PLATFORM DECKS AND HOLD.

There are two platform decks. On these decks are the magazines, storerooms, handling rooms, etc. On the upper platform are the 8-inch and 12-inch ammunition, forward fresh-water tanks, and one dynamo room at each end of the boiler compartments, and the steering gear and engine. On the lower platform are the 7-inch, 3-inch and 3-pdr. magazines and the forward and after torpedo rooms. In the hold are the chain lockers, 3-inch and 3-pdr., 1-pdr., 30-caliber and saluting-powder magazines, naval defense mines, torpedo war heads and after fresh-water tanks.

ARMOR.

The side belt is a complete water-line belt, 9 feet 3 inches wide amidships, for the length of the lower casemate; forward, this width is reduced to 8 feet and aft to 71 inches from frames 35 to 83, or 192 feet in wake of machinery spaces, the thickness is 11 inches, tapering from 11 inches at a point 5 feet 3 inches below top edge to 9 inches at lower edge, which is 5 feet below L. W. L.

Forward and aft of these points the thickness is 9 inches at top, tapering to 7 inches at bottom, then reduced to 7 inches at top and 5 inches at bottom, then to 5 inches constant thickness, then to 4 inches constant thickness.

Covering the middle length of the ship there is a lower casemate belt of 6-inch armor extending from the top of the heavy side belt to the lower edge of 7-inch gun ports, 2 feet 8 inches above the gun deck. Above this, and extending to the height of main deck, is a 7-inch belt, which is bent in to form wing plates at the 7-inch gun ports.

Between the gun deck and the protective deck, at frames 24 and 94, there are athwartship casemate bulkheads 6 inches thick; these are outside of the 12-inch barbette. Between the gun and main decks are similar 6-inch bulkheads at frames 28 and 86, connecting to the 12-inch barbettes. These athwartship bulkheads connect the ends of the casemate armor, forming a complete central casemate.

Between the gun deck and main deck, athwartship, splinter bulkheads of nickel-steel are provided for protection of 7-inch battery, each gun being thus in an isolated protected compartment. These are of 60-pound plates and located at frames 36, 50, 57, 64 and 78. Those in wake of boiler uptakes not being continuous. Those at frames 36 and 78 being continuous. A continuous fore-and-aft splinter bulkhead of 80-pound nickel-steel plate is fitted between 12-inch barbettes. Armored hinged doors are fitted in splinter bulkheads where necessary to provide access.

The 12-inch barbettes are 10 inches in thickness at front and 7½ inches thickness at back, from the top to the gun deck, and 6 inches between gun deck and protective deck. The 12-inch turrets are 12 inches thick on slope, remainder being 8-inch, top plate 2½ inches.

The 8-inch barbettes are 6 inches thick at front and 4 inches thick at back, with tubes from main to gun decks of 3¾ inches and from gun to protective deck of 3 inches thick.

The 8-inch turrets are 6½ inches thick on slope, remainder 6 inches; top plates 2 inches thick.

The conning tower is 9 inches thick and fitted with a swing door. The armored tube for communication is 6 inches thick. The signal tower aft is 6 inches thick.

BATTERY.

The 12-inch guns are mounted in pairs in two elliptical, electrically-controlled balanced turrets, each having an arc of fire of 270 degrees.

The 8-inch guns are mounted in pairs on four elliptical turrets, located two on each side and about equally spaced between the two 12-inch turrets. Each pair have an arc of fire of 145 degrees, the forward pairs from 90 degrees forward to 55 degrees abaft beam; after pairs, vice versa.

The 7-inch guns, twelve in number, are mounted singly on pedestal mounts with shields, in the central armored casemates on gun deck, each having an arc of fire of 110 degrees.

The secondary battery consists of twenty 3-inch R.F., ten 3-pdr. semi-automatic Hotchkiss, two 1-pdr. H. R. F. automatic, two 30-caliber Gatling machine guns. Four 21-inch submerged torpedo tubes are provided for, two forward and two aft. There are two Barr and Stroud range finders, one in each upper military top.

The following table gives the location and train of all guns, viz:

BATTERY.

4 12-inch B.L.R., 135 degrees port and starboard of center line of ship.
No. 1, forward turret, right hand.
2, forward turret, left hand.
3, after turret, right hand.
4, after turret, left hand.

8 8-inch B.L.R.
No. 1, forward starboard turret, right hand, 90 degrees forward, 55 degrees abaft beam.
2, forward starboard turret, left hand, 90 degrees forward, 55 degrees abaft beam.
3, forward port turret, right hand, 90 degrees forward, 55 degrees abaft beam.
4, forward port turret, left hand, 90 degrees forward, 55 degrees abaft beam.
5, after starboard turret, right hand, 90 degrees aft, 55 degrees forward beam.
6, after starboard turret, left hand, 90 degrees aft, 55 degrees forward beam.
7, after port turret, right hand, 90 degrees aft, 55 degrees forward beam.

No. 8, after port turret, left hand, 90 degrees aft, 55 degrees forward beam.

12 7-inch R.F., on gun deck.

Nos. 1 and 2, frame 31, 90 degrees forward, 26 degrees abaft beam.
3 and 4, frame 39, 73½ degrees forward, 42½ degrees abaft beam.
5 and 6, frame 53, 70 degrees forward, 46 degrees abaft beam.
7 and 8, frame 61, 46 degrees forward, 70 degrees abaft beam.
9 and 10, frame 75, 42½ degrees forward, 73½ degrees abaft beam.
11 and 12, frame 83, 26 degrees forward, 90 degrees abaft beam.

20 3-inch 50-caliber R.F.

Nos. 1 and 2, gun deck, frame 16½, 90 degrees forward, 30 degrees abaft beam.
3 and 4, gun deck, frame 101, 30 degrees forward, 90 degrees abaft beam.
5 and 6, gun deck, frame 109, 30 degrees forward, 90 degrees abaft beam.
7 and 8, main deck, frame 52½, 60 degrees forward, 60 degrees abaft beam.
9 and 10, main deck, frame 57, 60 degrees forward, 60 degrees abaft beam.
11 and 12, main deck, frame 61, 60 degrees forward, 60 degrees abaft beam.
13 and 14, upper deck, frame 36½, 90 degrees forward, 10 degrees abaft beam.
15 and 16, upper deck, frame 50, 90 degrees forward, 10 degrees abaft beam.
17 and 18, upper deck, frame 77½, 10 degrees forward, 90 degrees abaft beam.
19 and 20, forward bridge, frame 37, 10 degrees forward, 90 degrees abaft beam.

10 3-pounder semi-automatic Hotchkiss.

Nos. 1 and 2, upper deck, frame 39½, 45 degrees forward, 45 degrees abaft beam.
3 and 4, upper deck, frame 75, 45 degrees forward, 45 degrees abaft beam.
5 and 6, forward bridge, frame 40½, 45 degrees forward, 45 degrees abaft beam.
7 and 8, after bridge, frame 74½, 45 degrees forward, 45 degrees abaft beam.
9 and 10, after bridge, frame 77½, 45 degrees forward, 45 degrees abaft beam.
11 and 12, after bridge, frame 79, 45 degrees forward, 45 degrees abaft beam.

2 1-pounder H.R.F., automatic.

Nos. 1 and 2, upper deck, frame 64½.

2 machine guns, .30-caliber Gatling.

Nos. 1 and 2, forward bridge, frame 35.

4 torpedo tubes, 21 inches, 2 forward, 2 aft.

AMMUNITION SUPPLY.

Ammunition hoists are provided in number and speed designed to obtain as far as practicable a full supply of ammunition necessary for all guns firing at maximum speeds. In addition to the usual hoists in 8-inch and 12-inch turrets, there are twelve 7-inch hoists capable of handling either shell or charges, and fourteen combined 3-inch, 3-pdr. and 1-pdr. hoists. All hoists not within the armored citadel are protected with 80-pound nickel-steel plates. In order to supply ammunition to the hoists located along the ammunition passages on each side of lower platform deck, four continuous-motion ammunition-belt conveyors, each about 80 feet in length, are provided. These convey the shell and charges from the magazines located forward and abaft the machinery spaces on that deck.

These conveyors are so designed as to deliver ammunition to any of the hoists, the two forward ones supplying forward hoists, and two after ones the after hoists. They can also be operated in reverse direction to take ammunition into magazines. The shells are carried to the conveyor loading table from the shell room by means of trolleys on overhead rails. Charges and fixed ammunition are carried to conveyor by hand. The magazines for 8-inch and 12-inch ammunition are located on upper platform deck forward and abaft the machinery spaces. The 8-inch ammunition is carried from the magazines to base of turrets through the passages on differential blocks running on overhead trolleys. The 12-inch magazines are located adjacent to the handling rooms for those turrets.

The following table gives the number, location and type of the ammunition hoists, with the capacity of each.

All the 7-inch hoists combined can hoist all the 7-inch ammunition in about 41½ minutes.

Hoists 5, 6, 19 and 20, assisted by 23 and 24, can hoist all 3-pdr. and 1-pdr. ammunition in about eighteen minutes.

Hoists 11, 12, 15 and 16, and 19 and 20, assisted by 23 and 24, can hoist all 3-inch, 3-pdr. and 1-pdr. in about twenty-three and one-half minutes.

Mark of hoist.	Location at frame number.	Location side of ship.	Type of hoist.	Lift.	Approximate lift, in feet.	Number of guns to be supplied by each hoist	Total number of rounds.	Number of rounds in box or tank.	Total number of boxes.	Max. No. shells or boxes req'd to be hoisted by each hoist per minute to give specified rate of fire.	Bureau requirement No. of shells or boxes req'd to be hoisted by each hoist p. m.	Maximum capacity of hoist in boxes or shells per minute.	No. of rounds per m. for each hoist to deliver as per specifications.
4	32–33	P.	7-inch	low pl to gun deck	22½	1 7-in.	1,200	½ charge	2,400 tanks 1,200 shells	10½	7	7.04	2.33
3	32–33	St.	7-inch	Ditto	22½	1 7-in.	.....	.....	.....	10½	7	7.25	2.33
8	41–42	P.	7-inch	Ditto	22½	1 7-in.	.....	.....	.....	10½	7	7.00	2.33
7	41–42	St.	7-inch	Ditto	22½	1 7-in.	.....	.....	.....	10½	7	7.26	2.33
10	50–51	P.	7-inch	Ditto	22½	1 7-in.	.....	.....	.....	10½	7	7.26	2.33
9	50–51	St.	7-inch	Ditto	22½	1 7-in.	.....	.....	.....	10½	7	7.27	2.33
14	57–58	P.	7-inch	Ditto	22½	1 7-in.	.....	.....	.....	10½	7	7.16	2.33
13	57–58	St.	7-inch	Ditto	22½	1 7-in.	.....	.....	.....	10½	7	7.2	2.33
18	69–70	P.	7-inch	Ditto	22½	1 7-in.	.....	.....	.....	10½	7	7 11	2.33
17	69–70	St.	7-inch	Ditto	22½	1 7-in.	.....	.....	.....	10½	7	7.26	2.33
22	78–79	P.	7-inch	Ditto	22½	1 7-in.	.....	.....	.....	10½	7	7.40	2.33
21	78–79	St.	7-inch	Ditto	22½	1 7-in.	.....	.....	.....	10½	7	7.35	2.33
2	24–25	P.	3-inch	Hold to gun deck	30¾	1 3-in.	6,000 3-inch	4 3-in.	1,500 3-inch	3¾	7	7.10	2.8
1	24–25	St	3-inch	Ditto	30¾	1 3-in.	.....	.....	.....	3¾	7	7.45	2.8
6	35–36	P.	3″, 3 pdr & 1 pdr	Hold to upr deck	47	3 3-in 2 3-lb 1 1-lb auto.	7,200 3-lb. 5,640 1-lb.	16 3-lb 60 1-lb	450 3-lb. 94 1-lb.	16.67	7	6.82	6.32 12.60 42.00
5	35–36	St.	Ditto	Ditto	47	Ditto	.....	.....	.....	16.67	7	7.03	Ditto
12	55–56	P.	3″ to hoist, also 3 & 1 pdr	low pl main deck	31¼	1½, 3-in	.....	.....	.....	5.6	7	7.28	18.70
11	55–56	St.	Ditto	Ditto	31¼	1½, 3-in	.....	.....	.....	5.6	7	7 4	18.70
16	64–65	P.	Ditto	Ditto	31¼	1½, 3-in	.....	.....	.....	5.6	7	7.5	18.70
15	64–65	St.	Ditto	Ditto	31¼	1½, 3-in	.....	.....	.....	5.6	7	7 5	18.70
20	72–73	P.	3″ & 3 pdr	Berth deck to upr deck	23	1 3-in. 3 3-lb. 1 1-lb. 1 1-lb auto.	.....	.....	.....	11.55	7	8.19	3.10 17 85 18.20 60.60
19	72–73	St.	Ditto	Ditto	23	Ditto	.....	.....	.....	11.55	7	8.29	Ditto
24	89–90	P.	Ditto	h'ld to berth deck	23	to fe'd hoist U	.....	.....	.....	11.55	7	7 24	
23	89–90	St.	Ditto	Ditto	23	to fe'd hoist V	.....	.....	.....	11.55	7	7.38	
26	93–94	P.	3-inch	Hold to gun deck	29	2 3-in.	.....	.....	.....	7½	7	6.96	14
25	93–94	St.	3-inch	Ditto	29	2 3-in.	.....	.....	.....	7½	7	6.84	14
*7	43	P.	3″, 3 & 1 pdr Elec. whip	upper deck to bdg	8	1 3-in. 1 3-lb. 1 1-in auto 1 30-c colt	.....	.....	.....	7.3	...	Whip	...
*1	43	St.	Ditto	Ditto	8	Ditto	.....	.....	.....	7.3	...	Whip	...
*4	73–74	P.	3 pdr Elec. whip	Ditto	8	2 3-lb. 1 30-c colt	.....	.....	.....	3¾	...	Whip	...
*3	73–74	St.	Ditto	Ditto	8	Ditto	.....	.....	.....	3¾	...	Whip	...

*Whip hoists.

About 10 per cent. of 7-inch shells will be stowed near the guns.

Hoists 1 and 2, for 3-inch only. Hoists 5, 6, 19, 20, 23 and 24 will be assisted by hoists 11, 12, 15 and 16, the ammunition being whipped or passed from main to upper deck.

Hoists 25 and 26 to hoist 3-inch ammunition only.

WATER-TIGHT COMPARTMENTS.

There are 314 watertight compartments, of which 217 are below the protective deck, and 97 above the protective deck. They are distributed as follows, viz.:

On berth deck	28 compartments.
On protective deck	69 compartments.
On upper platform deck	50 compartments.
On lower platform deck	31 compartments.
In hold	67 compartments.
In inner bottom	69 compartments.

Compartments A92 to A99; B66 to B99; C89 to C99; D90 to D99, all inclusive, are double-bottom compartments, of which the following are wing compartments under coal bunkers and outboard sides of boiler rooms, viz: A86 to A93, B66 to B79, B88 to B93, C89 to C94, D90 to D95, all inclusive; they form the upper double bottom, between the armor and the 6th longitudinal, which is watertight. The remaining double bottom compartments are central compartments, extending from the 6th (watertight) longitudinal on one side to the 6th longitudinal on the other side of the keel, and non-watertight through the central line.

Double bottom compartments B94 to B99, inclusive, are fitted for reserve-feed tanks and compartments B88 to B93, inclusive, A1, A2, D10 and D11 are trimming tanks. C95 and C96 are under the engine rooms. The following tables give the capacities and the uses of various compartments fitted for special purposes, viz:

CAPACITIES OF DOUBLE-BOTTOM COMPARTMENTS.

Compt.	Gals.	Tons F. W.	Tons S. W.	Compt.	Gals.	Tons F. W.	Tons S. W.
A-97	14,677	54.5	56.05	B-90	5,359	19.9	20.45
A-98	18,041	67.0	68.88	B-91	5,359	19.9	20.45
A-99	24,234	90.0	92.54	B-92	4,471	16.6	17.05
B-80	19,441	72.2	74.28	B-93	4,471	16.6	17.05
B-81	16,614	61.7	63.48	C-95	23,991	89.1	91.68
B-82	23,588	87.6	90.08	C-96	19,926	74.0	76.08
B-83	20,384	75.7	77.85	C-97	19,899	73.9	75.97
B-84	17,071	63.4	65.20	C-98	19,360	71.9	73.97
B-85	24,018	89.2	91.74	C-99	18,714	69.5	71.51
B-86	20,681	76.8	78.97	D-96	18,256	67.8	69.74
B-87	17,234	64.0	65.82	D-97	17,017	63.2	65.00
B-88	6,248	23.2	23.88	D-98	17,502	65.0	66.82
B-89	6,248	23.2	23.88	D-99	17,754	66.3	68.27
	228,479	848.50	872.65		192,079	713.7	734.04

RESERVE FEED TANKS IN INNER BOTTOM.

B-94	5,804	21.55	22.20	B-97	4,982	18.50	19.06
B-95	5,804	21.55	22.20	B-98	4,151	15.42	15.88
B-96	4,982	18.50	19.06	B-99	4,151	15.42	15.88
					29,874	110.94	114.28

CAPACITIES OF TRIMMING TANKS.

Compt.	Gals.	Tons F. W.	Tons S. W.	Compt.	Gals.	Tons F. W.	Tons S. W.
A-1	4,907	18.22	18.74	D-10	14,848	55.13	56.71
A-2	7,764	28.83	29.65	D-11	10,352	38.44	39.54

CAPACITIES OF FRESH-WATER TANKS.

Tank.		Gals.	Tons.	Tank.		Gals.	Tons.
22-23 inboard,	1P. 1S.	1,517	5.66	98-99	1P. 1S.	2,040	7.61
21-22 middle,	1P. 1S.	937	3.50	98-99	C.L. of ship	1,868	6.97
21-22 outboard,	1P. 1S.	844	3.15	37-38	Gravity	800	2.98
22-23 outboard.	1P. 1S.	1,754	6.54	39-40	Gravity	800	2.98
				Total		10,560	39.39

COFFERDAMS.

	Cubic feet.		Cubic feet.		Cubic feet.
A-100	329	A-127	352	D-118	404
A-112	188	B-102	318	D-108	874
A-113	188	B-103	318		
A-114	289	C-104	342	*In Bunkers.*	
A-115	289	C-105	342		
A-116	390	C-106	288	B-104	316
A-117	390	C-107	288	B-105	316
A-118	389	D-109	296	B-106	341
A-119	389	D-110	296	B-107	341
A-120	383	D-111	375	B-108	370
A-121	383	D-112	375	B-109	370
A-122	370	D-113	394	B-110	370
A-123	370	D-114	394	B-111	370
A-124	302	D-115	635	B-112	398
A-125	302	D-116	635	B-113	398
A-126	352	D-117	404		

GENERAL NOTES.

Coaling Arrangements.—The coal capacity is 2,404 tons, of which 718 tons is carried above the protective deck and 364 tons in the transverse bunkers at the ends of the boiler space. This is handled by means of tripod coaling booms; eight of these coaling-booms are located on the main deck, four forward and four aft, and are arranged to lower coal bags through combination skylight and coaling hatches and companion and coaling hatches. Four coaling booms are located upper deck and are arranged to lower through combination skylight and coaling hatches and trunks. The coal is handled on the gun deck, where it is distributed to 24-inch scuttles and trunks leading to the various bunkers. The whips are operated by electric winches, two forward and two aft on main deck, and two on upper deck.

CAPACITIES OF COAL BUNKERS.

Lower Bunkers.

	Cu. ft.	Tons.		Cu. ft.	Tons.
B- 7	4,967	115.5	B-20	3,448	80.2
B- 8	5,619	130.7	B-21	3,317	77.2
B- 9	2,335	54.3	B-22	3,317	77.2
B-10	2,335	54.3	C- 5	2,152	50.2
B-11	2,290	53.3	C- 6	2,935	68.2
B-12	2,290	53.3	C- 7	2,244	52.2
B-13	3,102	72.2	C- 8	2,244	52.2
B-14	3,102	72.2	C- 9	2,529	58.8
B-15	3,437	79.9	C-10	2,529	58.8
B-16	3,437	79.9	C-11	2,300	53.5
B-17	3,455	80.4	C-12	2,175	50.6
B-18	3,455	80.4			
B-19	3,448	80.2	Below protective deck,		1,685.7

Above Protective Deck.

	Cu. Ft.		Tons.			Cu. Ft.		Tons.	
	La.	*Minn.*	*La.*	*Minn.*		*La.*	*Minn.*	*La.*	*Minn.*
B-118	2,650		61.6		B-126	3,238		75.3	
B-119	2,650		61.6		B-127	3,238		75.3	
B-120	2,978		69.3		B-128	3,302		76.8	
B-121	2,978		69.3		B-129	3,302		76.8	
B-122	3,273		76.1						
B-123	3,273		76.1					718.2	

Total, all bunkers, 2,403.9 tons.

Ventilation.—The ventilation of the ship is on the *plenum*, or supply system, except in the special cases of sick bays, waterclosets, dynamo rooms and steering-engine rooms. The ventilation blowers are electrically driven and the general

arrangement is in accordance with recent Navy practice; to avoid as far as possible the piercing of watertight bulkheads, the ventilating system is subdivided into a number of small systems. Forced ventilation is provided for all quarters, living spaces, passages, storerooms and magazines below the gun deck, also for all spaces over boilers and around magazines, and for all washrooms, closets and similar enclosures above the gun deck.

Fire Plugs.—There are 61 fire plugs in the ship, distributed as follows, viz:

Hold	5	Main deck	12
Lower platform	4	Upper deck	6
Upper platform	4	Bridge, for'd	1
Berth deck	12	Bridge, aft	1
Gun deck	16		—
			61

Masts and Spars.—There are two military masts. The fore mast is at frame 36, the main mast at frame 76. Each mast has an upper and a lower military top and a signal yard. The main mast has a lower searchlight platform, below the lower top. On the fore mast, just above the signal yard, is a lookout platform, fitted with guard rails and in communication with the bridge. The main mast is fitted with a small monkey gaff, and the wireless telegraph connections will be fitted on this mast.

Heights:
Above designed L.W.L., 24 feet 6 inches forward and aft.
24 feet 6 inches W.L. is 24 feet 4⅜ inches above base line.

Masts, fore and main:	
Truck light, top of, feet and inches	131- 9½
Ranging platform floor, top of (forward only), feet and inches	114- 2⅜
Searchlight platforms, feet and inches	75- 7⅛
Lower searchlight platform (aft only), feet and inches	42- 7⅛
Upper top, floor (forward only), feet and inches	74- 6⅛
Lower tops, floor, feet and inches	54- 7⅛
Bridge at 38 center line (forward), feet and inches	34- 9½
Flying bridge at at 38 center line, feet and inches	44- 7½
at side, feet and inches	43- 9⅛
After bridge at 74 center line, feet and inches	34- 6⅛
at side, feet and inches	33- 8⅛

Top of waist plate at stem, feet and inches	24– 0
beading at stern, feet and inches	20– 9½
main deck, planking at stem, feet and inches	20–10
stern, feet and inches	20– 4½
flat armor deck, feet and inches	3– 0

Power Watertight Doors.—There are 42 doors and 5 armored hatches, which are operated by electrical power. They are of the long arm system, similar to those installed on previous ships. They are so designed as to be operated on the spot by power or hand gear for either side of the bulkhead or deck. They can also be operated simultaneously by power from the emergency-control station located on forward bridge. Electrically-operated solenoid whistles are provided as a warning when doors are to be so closed. During any period of emergency operation, any individual door or hatch can be operated by power or hand, and after such operation during the period of emergency operation the closing repeats itself automatically.

Drainage and Flooding Systems.—The main drain, 15½ inches inside diameter, is fitted on starboard side between the inner bottom and boiler and engine-room floors and has a suction in each engine and boiler room above inner bottom; the valve for each suction is operated from compartment which it drains and also in deck plates on berth deck. The main drain is connected directly with the main circulatng pumps, and has an auxiliary connection to the construction and repair drainage manifold near each fire and bilge and auxiliary feed pump. These drainage manifolds are connected to steam engineering pump manifolds.

The secondary drain, 5½ inches diameter, is run on port side throughout length of machinery spaces and is connected to each construction and repair drainage manifold above mentioned. Branches are led from the secondary drain to bilge wells located in after ends of boiler and engine compartments, those on port side leading directly into drain, and those on starboard side leading through the manifold suctions to bilges forward and abaft machinery spaces, to shaft alleys and double-

bottom compartments are lead to the drainage manifolds. All valves on secondary drain are operated 30 inches above floor plates.

Double-bottom compartments under machinery spaces, except those used for reserve-feed water, are provided with means both for drainage and flooding; outside these spaces with means for drainage only, except for those outboard of sixth longitudinal, where piping is to be fitted for either drainage or flooding. In general, compartments not provided with other means of drainage will be drained by means of hose led through doors or hatches.

The fire mains are of 6-inch copper pipe, sabined and run continuously through machinery spaces on each side of ship, directly under protective deck beams. The port and starboard systems are cross connected at ends of machinery spaces, forming a circuit to which all fire and bilge pumps are connected direct. Risers from main are fitted to supply necessary plugs in each compartment above. Forward and aft of machinery spaces throughout magazine spaces, a single main is led from the cross connection to supply risers and plugs in those localities. The fire main is also connected to the magazine-flooding system on upper platform. It has also by-pass connections to the flushing system, the full diameter of the latter.

The flushing system is 4-inch copper pipe, sabined and installed under gun-deck beams. This supplies salt water to all spaces throughout ship where required. On upper platform, forward, two separate motor-driven centrifugal pumps are provided for flushing and providing salt water for showers, washrooms, heads, etc.

The fresh-water service throughout the ship is provided by means of a gravity tank located on upper deck and of 1,600 gallons capacity. Two electrically-operated, reciprocating, fresh-water pumps are provided, one aft and one forward, for general service of all tanks.

Boats.—The following table gives the number and sizes of the boats, with location of each and weight carried by each.

Small boats.	Men.		Water and provisions.		Weight of each boat, stores and men.	Location on bridge deck.	Frames.
	Each.	Total.	Each.	Total.			
					Tons.		
1 50-ft. steam cutter	...	60	.....	600	19,680	Starboard aft.....	52–63
(40-ft. steam cutter temporarily fitted.)							
2 36-ft. steam cutters	53	106	530	1,060	16,961 each	P. & S. aft.....	53–62
1 36-ft. launch	...	78	.....	780	9,953½	Port aft.....	53–62
3 33-ft. launches	64	192	640	1,920	8,470½ each	1 Starb. aft..... 2 Port aft.....	54–63 53–63
5 30-ft. cutters*	45	225	450	2,250	5,624½ each	1 Starb. aft..... 2 Port aft.....	54–62 54–62
1 30-ft. barge (on davits, main deck)	...	37	.....	370	5,200	Starboard aft.....	96–103
2 30-ft. whale boats (on davits, main deck)	29	56	280	560	4,802½ each	1 Port aft.... 1 Starb. aft..	76–84
1 30-ft. whale boat (gig, main deck)	...	20	.....	200	4,471½	Port aft.....	96–103
2 20-ft. dinghies	12	24	120	240	2,079½ each	1 Starb. aft.. 1 Port aft....	65–71
1 16-ft. dinghy	...	5	.....	50	1,324½	Port aft.....	65–71
1 14-ft. dinghy	...	4	.....	40	1,029½	Starboard aft.....	65–71

* The sea position for two of these cutters is on davits, main deck, between frames 30 and 38.

STEERING ENGINE.

The steering gear is operated by a Williamson Brothers' steam-steering engine, operated through a right-and-left-hand screw by a traveler with side rods to the crosshead. The contract requires that the rudder can be put from hard over to hard over, 70 degrees in all, in 20 seconds, with ship at full speed. The operating steam pressure is 150 pounds. The valves are worked by wire-rope transmission from flying bridge, bridge, conning tower and communication room. Valves may also be operated by a handwheel at engine. There is a hand-steering wheel in steering-engine room, and an arrangement for relieving tackles direct to crosshead.

During trial trip and after completion of the four-hours' speed trial, the vessel being still operated under forced draft, tests of the steering gear were made as to its fulfilling the contract requirements. The helm being amidships, it was put hard-a-port, the time required being 11 seconds and the time required for the vessel to swing through slightly over 360 degrees, was 4 minutes and 18 seconds. The helm was then put from hard-a-port to hard-a-starboard, the time required being 18 seconds, and the time required for the vessel to swing through approximately 360 degrees during this second circle was 6 minutes, 46 seconds. On completing these tests the helm was put amidships, and after the fires had been sufficiently reduced to keep steam pressure under control, and while vessel was going astern at a rate of about 60 revolutions per minute of the main engines, the steering gear was tested by bringing the helm to various positions up to the hull helm angles, while the vessel maintained a course astern. The steering gear operated in an entirely satisfactory manner during the above trial.

ANCHOR WINDLASS.

There is one Hyde Windlass Co.'s steam windlass with four wildcats, capable of hoisting two 17,600-pound anchors at the same time. It is of the worm-gear type, operated by a vertical shaft, which is connected to the engine

on gun deck by bevel gears. The engine is designed to work at 150 pounds steam pressure.

During the trial trip and after completion of standardization, trial tests of the anchor gear were conducted as follows: The starboard anchor was dropped and chain veered until the 60-fathom shackle was at the water's edge, the vessel being in about 30 fathoms of water. She was then given a slight sternboard so as to have starboard anchor chain tend on the starboard bow, when the port anchor was dropped and both chains veered until starboard chain was veered to bitter end and the 60-foot shackle on port chain was at the waterline. The blocks of both controllers were then dropped and at 12 h. 12 m. windlass was started heaving in on both chains. At first, chain was taken in at about 8 fathoms per minute and this speed was maintained with slight variations throughout the test. After 8 minutes the port anchor was up, and after stoppering that chain and disconnecting its wildcat the windlass was again started, at 12 h. 23 m., raising starboard anchor, which was up at 12 h. 29 m. The windlass worked in general in a very satisfactory manner. It was noted that, due to cold weather, the lubricating lard oil was thickened and the pumps handled it with difficulty. The Board recommended that a heating pipe be fitted for the oil used for worm and worm wheel, as an abundant flood of oil is considered essential to their satisfactory operation.

The Board reported finally as follows:

First.—The *Louisiana* maintained a true average speed of 18.82 knots for a period of four hours.

Second.—The steering qualities, steadiness and seaworthiness of the vessel are excellent. (Subject to result of test of final inclining experiments.)

ELECTRIC PLANT.

The electric plant is of a greater capacity than any other that has been installed, so far, in any other vessel built for the U. S. Navy. There are eight 100-kilowatt generating sets, of 125 volts pressure at the terminals; the generators

were supplied by the General Electric Co., and are of the direct-current, compound-wound, multipolar type. The wiring is on the two-wire system.

These eight machines are placed in two dynamo rooms, four in each room. The dynamo rooms are situated under the protective deck, one just forward of the boiler compartments, the other just abaft the boiler compartments, on the upper platform. Steam is supplied to the generators directly from the auxiliary steam pipe; and two dynamo condensers, one in a pocket just under each dynamo room, are provided for taking the exhaust steam of only the dynamos; these condensers are so connected that the dynamo exhaust, forward or aft, may be led into either dynamo condenser or into either auxiliary condenser in the engine room.

Dynamo Engines.—The engines are of the vertical, cross-compound, General Electric Co.'s type; all the working parts are enclosed and lubricated under pressure; the introduction of oil into the cylinders is guarded against by fitting a top plate on the enclosing covering, in which are soft-packed stuffing boxes for the rods and valve stems to work through.

Dynamo engines, 100-kilowatt, number of	8
Diameter of cylinders, H.P., inches	10
L.P., inches	18
Stroke, inches	10
Diameter of piston-rods, H.P., inches	2.25
L.P., inches	2.25
Revolutions per minute, full load	350

Motors.—All the auxiliary machinery outside the engine and firerooms, except the ice machine, the anchor engine, the forced-draft blowers and the steering engine and evaporator pumps, is driven by electric motors. The following is a list of the motors installed, viz:

Number.	H.P.	Winding.	Use.
42	1	Comp.	For operating power watertight doors.
5	1	Comp.	For operating power watertight hatches.
2	50	Series	Hoisting motors for boat cranes.
2	30	Series	Rotating motors for boat cranes.
6	30	Comp.	For deck winches.
26	3	Shunt	For endless-chain ammunition hoists.

Number.	H.P.	Winding.	
4	3.5	Shunt	For ammunition whips.
4	4	Shunt	For endless-chain ammunition conveyors.
4	25	Shunt	For turning 12-inch turrets.
8	15	Shunt	For turning 8-inch turrets.
32	Var.	.....	For ventilating fans.
1	6	Shunt	For operating steam-laundry machinery.
2	2	Shunt	For operating fresh-water pumps.
2	5	Shunt	For operating flushing pumps for head.
*2	1.5	Shunt	For drainage pump for firemen's washrooms.

In addition to the above there are the following:

6 Portable ventilating sets of ¼ H.P. each.

45 Desk-and-bracket fans of $\frac{1}{12}$ H.P. each.

8 Bracket fans, of $\frac{1}{6}$ H.P. each.

Motor-Generators.—Turret Control System.—The turrets are controlled on the Ward-Leonard system. The current for driving the turret-training motors and the gun-elevating motors, both 12-inch and 8-inch, is furnished by motor generators in each turret, which, in turn, are driven by current from the main generators. With this arrangement one main generator may be used to furnish current for both turret power and lighting at the same time, without having the flickering of lights due to variation in the load when working the turrets.

For each 12-inch turret there is provided a 25-kilowatt motor generator for training and for each 12-inch gun, a 6.4-H.P. motor generator for elevating. For each 8-inch turret, there is provided a 15-kilowatt motor generator for training, and for each 8-inch gun, a 4.7-H.P. motor generator for elevating.

Speed regulation is obtained by varying the field of the motor generator up to a certain limit, and beyond that by varying the field of the training or elevating motor.

Lighting Output.—The output for lighting supplies 1,100 16-candle-power, incandescent lamps; an additional 100 lamps will probably be provided; also 10 3-ampère arc lights; two 60-inch searchlights (of about 42,000 candlepower each)

*Not yet approved.

and four 30-inch searchlights (of about 20,000 candlepower each); two night-signalling sets, two truck lights; two diving lanterns, with eight 150 candlepower lamps.

System of Distribution.—The wiring is on the 2-wire feeder system. In each dynamo room, there is a generator switchboard to which the four machines may be connected. On the opposite side of the dynamo room, in a separate watertight compartment, there is a distribution board. The forward distribution board feeds all circuits leading to outlets forward of the amidship section of the ship; the after distribution board feeds all circuits leading to outlets abaft the amidship section.

Either distribution board may be connected with either generator switchboard; thus, any generator may be connected with any circuit, lighting, power or turret.

The following circuits are provided with feeders from each distribution board and with double-pole throw-over switches, allowing the use of either feeder; this duplicates the feeders to these circuits viz:

All lighting circuits in firerooms.

All lighting circuits in engine rooms.

Each circuit feeding motor generators.

Each circuit feeding turret-ammunition hoist and training motors and miscellaneous turret power.

Turret-hoist Indicators.—A 16-candlepower incandescent-light signal is provided for signalling from the handling room when the ammunition-lift car is loaded and ready for hoisting.

A 16-candlepower, green, incandescent-light signal, automatic in operation, is provided for indicating when the car of the turret-ammunition lift arrives within 42 inches of the extreme lower position.

Interior Communication.—A complete central-station system of telephones is installed. In addition, private lines are run for executive officer, navigator, chief engineer, surgeon and electrical gunner. There are twenty-five separate lines from the central station and seven private lines. In addition, there are eighty-seven lines of voice pipes.

The telephones are a special type of loud-speaking instrument, made by the Cory Company. With these instruments four wires are required for each instrument, with a common return wire connected to all the call bells and buzzers, instead of six wires, as required by the Bell system. Those instruments located in stations where they will be subject to exposure, moisture or rough usage, or where there is a great amount of noise, are of watertight type. There are two ear pieces which encircle the ear and exclude external sounds. The instruments are said to work very satisfactorily, even in exposed places in a high wind.

Warning Signals for Watertight Doors.—A system of whistles, operated by solenoids, is installed to give warning of the closing of the power watertight doors.

All feeders and wires are led through the usual heavy and expensive pipe-conduits, by which about 18 tons is added to the weight of the ship.

MAIN ENGINES.

There are two four-cylinder triple-expansion engines of the vertical, inverted-cylinder, direct-acting type, placed abreast each other in separate watertight compartments, which communicate through a watertight door in the centerline bulkhead. The engines are of the contractors' design, built under specifications of the Bureau of Steam Engineering. The engines are right-and-left, turning inwards when going ahead. The engines were designed to develop 16,500 horsepower when making about 120 revolutions (for a speed of 18 knots) per minute. On the standardization trial, the horsepower at 120 revolutions was 16,128. The cylinder diameters are: H.P., 32½; M.P., 53; both L.P., 61.

The arrangement of the cylinders, beginning forward, is: forward L.P., H.P., I.P., after L.P. The arrangement of cranks is: the forward L.P. and H.P. cranks are opposite; also the M.P. and the after L.P. cranks, the second pair being at right angles with the first. The engines turn inwards going ahead, and the sequence of the cranks is: H.P., M.P.,

forward L.P., after L.P. The crank shaft is in two sections.

The engine framing is of the Bailey design, the cylinders of each engine being supported by six pairs of forged-steel columns which are tied together and braced by forged-steel stays. This is an admirable design, very rigid and strong and leaving the moving parts accessible for oiling, cleaning and overhauling. The forward L.P. and H.P. engines are secured together, as are the after L.P. and the M.P. The pairs of cylinders of one engine are secured across to those of the other engine by tie rods through the middle-line bulkhead, and by tie rods to the forward and after engine-room bulkheads. The vibration on trial was inappreciable.

The method of securing the crosshead guides is different from that in the usual design, and is as follows: To the top of the engine columns a cast-steel strong back, or girder, of I-section is bolted, its axis horizontal; another similar girder is bolted to the columns at their mid-height. On each girder, between each pair of columns, are four swelled bosses, through which are drilled bolt holes; through these holes pass the bolts which secure the crosshead guides to the girders. The guides are further supported or hung by short tie rods, between the top of each guide and the bottom of the cylinder.

The engine bed plates are of cast steel, and are made in three sections for each engine, bolted together at flanges. Each section carries two main bearings.

The longitudinal and cross portions of the bed plates are of I-section.

The bed plates are secured to the keelson plates worked in the hull. Seatings for engine columns, bearing brasses and caps, for turning engine, etc., are provided.

The stem of the throttle valves is carried down so far that the throttle wheel falls below the height of a man's waist, with the result that the wheel must be turned to the left to close the valve. This arrangement would probably result in fatal confusion at a time of excitement or emergency, and the wheel will be arranged to turn to the right to close.

The H.P. cylinder has one piston valve; each of the other

cylinders has two piston valves, all equal in diameter; the L.P. and M.P. valves differing only in length.

The valve gear is of the Stephenson type, with double-bar links, working directly and fitted with adjustable cut-off blocks, by which the cut-off may be adjusted between the limits of 0.5 and 0.76 of the stroke.

No starting valves are provided, arrangements being provided for admitting steam directly to the receivers for starting the engines.

All cylinders are jacketed around the barrel and at both ends, with the exception of the H.P., which is not jacketed at the ends. All jacket steam passes successively through the H.P., M.P. and L.P. jackets, the route being as follows: From the bottom of H.P. jacket to the top of I.P. jacket, and from bottom of I.P. jacket to tops of L.P. jackets, reducing valves being fitted between. Each jacket has an independent drain.

Each engine has an 11-inch balanced throttle valve. The throttle is balanced by a balance piston, the diameter of which is equal to that of the disc of the throttle, live steam passing through a 1-inch by-pass to the top of this balance piston.

There is also a 2-inch by-pass on the throttle for use in warming up, or in making slight variations of speed, etc.

Each main engine has a reversing engine of the usual floating-lever, oil-controlled type. For each main engine there is a turning engine, which is double and is fitted with link motion.

The arrangement of the upper engine-room hatch (in the central part of which is located the work shop) is such as to give ample room for taking off cylinder covers, for which there is a very good arrangement of overhead trolley tackles.

DATA FOR CYLINDERS.

Cylinders, No. for each engine	4
H.P., diameter inches	32½
I.P., diameter, inches	53
L.P., diameter, inches	61
Volume swept by H.P., piston, mean, per stroke, cubic feet	22.470
I.P. piston, mean, per stroke, cubic feet	60.709
L.P. piston, mean, per stroke, cubic feet	80.604

Ratio, net area, I.P. piston to H.P. piston	2.70
L.P. piston to H.P. piston	7.17
Cylinder walls, thickness, H.P., I.P., L.P., inches	1⅜, 1 7/16, 1 5/16
liners, thickness, H.P., I.P., L.P., inches	1½, 1 5/16, 1½
Valve-chest liners, thickness, H.P., I.P., L.P. inches	1½, 1½, 1½
Jacket space, inch	⅝
Cylinder relief valves, diameter (2 each cylinder), inches	3½
Valve-chest relief valves, diameter, inches	4½

Pistons.—The pistons are of the usual dished or conical form. The H.P. piston is of cast iron and comparatively heavy for balance; the M.P. and L.P. pistons are of cast steel.

The piston rings are of cast iron. The H.P. and M.P. have each one ring, with two packing grooves; at the point where each ring is cut there is, on the inside of each end of the ring, a square shoulder ⅜ inch high by 1 inch wide; over these shoulders is fitted or mortised a steel tie piece, uniting the ends. Thus the ring is practically a solid ring, adjustable for wear.

The L.P. has two rings, each in 6 sections, jointed with the usual tongue piece.

The followers are all of forged steel.

A thread is turned on the upper end of each piston, for an inch and a half, to receive a nut for use in handling the piston.

DATA FOR PISTONS.

Diameters of pistons, H.P., M.P., L.P., inches	32½, 53, 61
Width of bearing surface of packing rings, inches	2½
Thickness of packing ring, inch	1
Number of follower studs, H.P., I.P., L.P	10, 18, 22
Diameter of follower studs, inches	1½

Piston rods.—The piston rods are of high-grade forgings; the ends are tapered in the usual manner to fit into the pistons and crossheads.

Diameter of piston rods, inches	7½
hole in rods, H.P., I.P., L.P., inches	1½, 3, 4⅝

Crossheads.—Crossheads are of class "A" forgings. The crosshead pins have tapered axial holes, and the surface of the pins is flattened on each side, reducing the bearing at the sides of the brasses and leaving a space for lubricant.

Length of crosshead pins, inches	9¼
Diameter of crosshead pins, inches	9¼
axial hole in pins, inches	from 6 to 4

Crosshead Slipper.—The slippers are of cast steel; they are secured to the crosshead by six 1½-inch tap bolts, and are lined with white metal in the usual way.

Length of crosshead slippers, inches	27½
Width of crosshead slippers, inches	21
Length of backing surface, inches	27½
Width of backing surface (each side), inches	7¾

Crosshead Guides.—The ahead guides are of cast iron, bolted at top and bottom to the strong backs on the engine columns, as described before, and stiffened by webs on the back. The backing guides are of cast steel, bolted to the ahead guides, with liners between to permit adjustment for wear.

PROPELLERS.

There are two three-blade propellers; both are inward turning for ahead motion.

The blades and the hub are of manganese bronze. The hub is fitted on the tapered end of the shaft and held in place by keys and a nut in the usual manner. The blades fit in recesses in the hub, and the boltholes are elongated to permit adjusting the pitch of the blade.

The design of the propellers for the *Louisiana* differs in some essential points from that of the sister ship, the *Connecticut*, as may be seen from the following comparative table, viz:

	Louisiana.	*Connecticut.*
Diameter, feet and inches	17- 3	17- 9
Pitch, feet and inches	18	17- 6
Number of blades	3	3
Helicoidal area, square feet	84.5	99
Projected area, square feet	71.5	84.3
Disc area, square feet	234	247
Ratio, projected to disc area	0.305	0.341
Area of immersed midship section, square feet	1,808	1,808
Ratio, disc area to I.M.S	0.1294	0.1366
projected area to I.M.S	0.0395	0.046

DATA FOR *LOUISIANA'S* PROPELLERS.

Immersion, upper tip of blade, feet and inches		6- 1.5
Lower tip of blade, above keel, feet and inches		0- 4.2
Average revolutions per minute, official trial	S.	127.66
	P.	127.69
Slip in per cent. of its own speed	S.	16.99
	P.	17.01
Number of tap bolts for blades (7 each blade)		42
Diameter of tap bolts, inches		4
Length, under head of tap bolts for blades, inches.		8
Width of longitudinal keys (2 keys), inches		2½
Thickness of longitudinal keys, inches		1½
Diameter of nut thread, inches		11
Length of nut thread (4 to 1 inch), inches		7

After casting, the blades were ground, tinned and burnished. The propellers were swung between centers, on a mandrel, and accurately balanced by removing metal from heavier blades.

SHAFTING AND SHAFT BEARINGS.

The shafting, of class "A" forgings, is in four sections, viz: thrust, line, stern-tube and propeller. The stern-tube section is coupled to the line shaft so as to permit withdrawal through the stern-tube, as follows, viz: 1. Inboard coupling: About one foot of the forward end of the stern-tube shaft is increased in diameter by about two inches, and on this swelled end is keyed a steel ring or collar, which forms a detachable flange; another, thinner ring, in halves, is fitted abaft the first flange ring and the whole is through bolted to the flange on the after end of the line shaft. 2. Outboard coupling: The after end of the stern-tube shaft and the forward end of the propeller shaft are tapered for a length of about 32 inches; these tapered ends fit into a forged sleeve, taperbored to correspond, to which the shaft ends are keyed, each end by two longitudinal steel keys and one through key or cotter, thus forming the coupling.

The sections of the composition casing around the stern-tube shaft are joined by a rabbet joint; at the middle point of the lap of the rabbet, a $\frac{3}{16}$ groove (cut half into one section, half into the other) is cut and afterwards pumped full

of red lead to seal the joint. The stern tube is made tight at its forward end by the usual stuffing box and gland, having a packing space 7½ inches in depth and 1 inch in width, with longitudinal scores to prevent rotation of packng. Abaft the packing space is fitted a 1½-inch drain pipe, connected to water-service pump and engine-room bilge.

DATA FOR SHAFTING (ALL DIMENSIONS IN INCHES).

Diameter of thrust shaft	15¼
hole	9¼
Number of thrust collars	13
Outside diameter of thrust collars	24¼
Width of thrust collars	2
space between thrust collars	4
Diameter of line shaft	15¼
hole	9¼
stern-tube shaft	18¼
hole	9¼
Length of stern-tube bearing, forward	54
aft	54
Diameter of propeller shaft	16¼
hole	9¼
coupling flange	30
Thickness of coupling flange	3¼
Number of coupling bolts	8
Diameter of coupling bolts	3¼
pitch circle for coupling bolts	23¼
Width of longitudinal keys	2¼
Depth of longitudinal keys	1¼
Width of through keys, top	7½
bottom	6½
Thickness of through keys	2

Bearings.—There are six *main* bearings for each crank shaft, two *steady* bearings for each thrust shaft, two *spring* bearings for each line shaft, two *stern-tube* bearings for each stern-tube shaft, and one *strut* bearing for each propeller shaft.

Thrust bearings.—The thrust bearings are of the usual horse-shoe pattern, with a steady bearing at each end. The pedestal is of cast-iron; it is bolted to a cast-iron sole plate, which is riveted to the foundation built up from the hull. The sole plate has raised lugs at the ends, with machined faces; to adjust the pedestal in a fore-and-aft direction steel wedges

are fitted between these lugs and the end of the pedestal, the bolt-holes in the pedestal being oblong.

The horse shoes or thrust shoes are of cast steel, each made hollow for circulation of water through it and faced on each side with white metal. Each thrust shoe is independently adjustable by means of wrought-iron nuts on each side on screwed side rods. The sides and ends of the pedestal form a trough at the bottom for oil, in which the thrust collars turn. In the bottom of the trough is a closed coil for circulating cooling water. A stuffing box is fitted at each end of the pedestal casting to prevent escape of oil.

No suitable movable cover is provided, and much dirt finds its way into the trough.

Number of thrust shoes	12
Width white-metal bearing surface, inches	4½
Diameter of thrust side rods, inches	3¾
Length of each steady bearing, inches	18
spring bearing, inches	24½
stern-tube bearing, forward, inches	54
aft, inches	54
strut bearing	51

REVERSING ENGINE.

Each main engine is provided with a steam-reversing engine having an oil-control cylinder. The reversing engine is located on the inboard side of the high-pressure cylinder casing, and is worked by the usual floating-valve gear from the working platform. A hand oil pump is provided for use when hand reversing is necessary.

DATA FOR ONE ENGINE.

Diameter of steam cylinder, inches	15
Diameter of oil cylinder, inches	7½
Stroke, inches	20

TURNING ENGINE.

The turning engine is located abaft the after L.P. engine column on the inboard side of each engine, and is a double engine fitted with link motion for reversing.

DATA FOR ONE ENGINE.

Number of cylinders	2
Diameter of cylinders, inches	7
piston rods, inches	1⅝
Stroke, inches	5
Angle of cranks, degrees	90

MAIN CONDENSER.

One main condenser is located on the outboard side of each engine room. The shell is made of boiler plate, the heads of composition and the tube sheets of Muntz metal. The tubes are not tinned, and both tubes and ferrules are made of Admiralty metal. The forward head of each condenser is fitted with a "butterfly" valve, so that the water from the circulating pump can be passed directly overboard without going through the condenser tubes. Zinc plates are placed inside on the manhole plates of each head. The shell is lagged with cowhair felt and covered with galvanized sheet steel.

The following are the data for one condenser:

Diameter of shell, feet and inches	6- 7
Thickness of shell, inch	⅜
Length over all, feet and inches	15- 5½
Tubes, outside diameter, inch	⅝
length between tube sheets, feet and inches	13- 0
total length, feet and inches	13- 2½
thickness, No. 18, B.W.G., inches	.049
spacing between centers, inch	1⅛
number in each condenser	4,878
Cooling surface, each condenser, square feet	10,375

PUMPS.

Main Air Pumps.—The main air pumps are of the Blake, twin, vertical-beam type, single acting, with two steam and two water cylinders. The foot, bucket and discharge valves are of uniform size throughout, and each is composed of three discs, the upper disc having a single hole for valve stud or spindle, while each of the lower ones has in addition eight ¼-inch holes drilled near its periphery in such a manner that the disc above will cover the holes in the disc below when the

valve as a whole is seated, but will allow the escape of water between the discs.

The foot, bucket and discharge valves may be examined and removed through suitably located handhole plates, but only the discharge valve seats are removable.

The piston rods are made of nickel-steel, the pump rods of Tobin bronze, and the water barrels, heads and buckets are of composition; buckets packed with fibrous packing.

The following are the data for one pump:

Diameter of steam cylinders (2), inches	12
pump cylinders (2), inches	30
Stroke, inches	18
Diameter of piston rods (nickel-steel), inches	2¼
pump rods (Tobin bronze), inches	2¾
valves, bucket, head and foot, inches	6
Number of valves (10 each in bucket, head and foot) in each barrel	30
one complete pump, both barrels	60
Area through two valve openings, square inches	162
Lift of valves possible under guards, square inches	1⅛

TRIAL DATA FOR AIR PUMPS.

	Starboard	Port.
Double strokes per minute	21.5	21.9
Vacuum, inches	26.5	26.5
Indicated horsepower, each pump	12.25	12.62

Main Circulating Pumps.—For each main engine, there is one double-inlet, centrifugal circulatng pump, driven by an independent, two-cylinder, compound, vertical, inverted engine, with cranks at right angles. The pumps are designed to deliver each 12,000 gallons of water per minute from the bilge at a speed not exceeding 265 revolutions per minute. The pumps are located outboard, just forward of the main condenser.

Each pump is fitted to draw from the sea, the main drain and the engine-room bilge, and to discharge through the condenser overboard or directly overboard, through valves in the condenser bonnet (a butterfly valve in the forward bonnet). The three suction valves are interlocked so that the main injection valve, or the bilge-injection and main-drain valve, only can be opened at the same time, a departure from the old,

usual arrangement by which no two of these valves could be opened at the same time.

The casing and runner of the pump are made of composition and the shaft of phosphor-bronze.

DATA FOR ONE PUMP AND ENGINE.

Steam cylinders, diameter, H.P. and L.P., inches	11 and 22
Stroke, inches	10
Pump runner, diameter, inches	45
width at periphery, inches	4
hub, inches	$10\frac{9}{16}$
Inlet nozzle, diameter, inches	$18\frac{1}{2}$
Discharge nozzle, diameter, inches	21

TRIAL DATA FOR MAIN CIRCULATING PUMPS.

	Starboard.	Port.
Revolutions per minute	162.3	148.8
Indicated horsepower	48.5	38

Auxiliary Air and Circulating Pumps.—Dynamo Air and Circulating Pumps.—In each engine room there is one Blake, simplex, horizontal, combined air and circulating pump; in the pocket under each dynamo room, there is a similar, but larger dynamo air and circulating pump. Each air pump has a suction from its own condenser and a discharge to the main feed tanks; each circulating pump has a suction from and a discharge to the sea.

DATA FOR AUXILIARY, AIR AND CIRCULATING PUMPS.

	Engine room.	Dynamo room.
Steam cylinder, diameter, inches	6	10
Air-pump cylinder, diameter, inches	10	14
Circulating-pump piston, diameter, inches	10	16
Stroke, inches	12	12
Valves, circulating pump, diameter, inches	$5\frac{5}{8}$	$3\frac{1}{2}$
air pump, diameter, inches	$5\frac{5}{8}$	7
circulating pump, number	8	44
area through one, square inches	13.6	5
air pump, number	8	4

TRIAL DATA FOR ONE AUXILIARY AND CIRCULATING PUMP (STARBOARD).

Double strokes per minute	30
I.H.P.	19

Main Feed Pumps.—There are four main feed pumps, two in each engine room. They are of the Blake, vertical, simplex, double-acting type, with outside center-packed plungers, sucking from the pipe connecting the main feed tanks and discharging through a grease extractor into the main feed discharge, directly or through the feed-water heaters.

Piston rods are of nickel-steel. Pump rods are of Tobin bronze. Valves are of phosphor-bronze held on their seats by phosphor-bronze springs.

A zinc protecting ring is fitted on each valve guard plate. A sheet-brass dust guard, easily removable, is fitted around the water cylinder to protect the exposed part of the plunger.

Each pump has a relief valve, the intention being to keep the pumps running steadily, the surplus water going through the relief valve.

DATA FOR PUMPS.

Steam cylinder, diameter, inches	13
Water cylinder, diameter, inches	9½
Stroke, inches	12
Water valves, diameter, inches	3
number in one pump	20
Piston rod, diameter, inches	2½
Pump rod, diameter, inches	3

TRIAL DATA FOR MAIN FEED PUMPS.

	Starboard.	Port.
Double strokes per minute	28.25	30.25
Indicated horsepower	47.25	49.6

Hot-Well Pump.—There are two Blake, vertical, simplex, hot-well pumps, one in each engine room, taking water from the suction of either starboard or port air pump, from the hose connections outside on the ship's side and from the reserve-feed tank, and discharging into main-feed tanks or reserve-feed tanks. Piston rods are of nickel-steel. Pump rods are of Tobin bronze. Valves are of hard rubber. Plungers are packed with rings of lignum vitæ. The valve seats are tapered composition rings, driven into the casting.

DATA FOR PUMPS.

Steam cylinder, diameter, inches	12
Water cylinder, diameter, inches	14
Stroke, inches	12
Steam piston rods, diameter, inches	2¼
Pump rods, diameter, inches	2¼
Number of water valves	36
Diameter of water valves, inches	3½
Area through one valve seat, square inches	5

Distiller Circulating Pump.—There is one Blake, vertical, simplex, double-acting, distiller circulating pump, located in the starboard engine room, with a suction from the sea and discharge to the distillers and flushing system. This pump is in every respect similar and of equal dimensions to the hot-well pump except that zinc rings are fitted around the valve seats for protection.

Auxiliary Feed Pumps.—There are six Blake, vertical, simplex, double-acting, outside-center-packed plunger, auxiliary feed pumps, one in each boiler room; each with a suction from the feed-tank connecting pipe, the main drain, secondary drain, fire-room bilge, sea, bottom blows in same compartment, to the fresh-water manifolds and suction-hose connection in the same compartment, and discharging into the auxiliary feed line, fire-main, overboard in the same compartment and to a hose connection. The details of the pump are similar to those of main feed pumps.

DATA FOR ONE PUMP.

Steam cylinder, diameter, inches	11
Water cylinder, diameter, inches	8
Stroke, inches	12
Piston rod (nickel-steel), diameter, inches	2¼
Pump rod (Tobin bronze), diameter, inches	2¼
Valves, number in each pump	20
diameter of each, inches	2¾
Area through one, square inches	3.24

Fire and Bilge Pumps.—There are two Blake, vertical, simplex, double-acting fire and bilge pumps, one in each engine room. Each pump has a suction from the sea, the main drain, the secondary drain, the drainage manifold, the engine-room bilge, the crank pits and a suction hose connection; each pump

has a discharge into the fire main overboard and through a hose connection. The details of the pump are similar to those of the distiller circulating pump. The valves are of medium rubber.

DATA FOR ONE PUMP.

Steam cylinder, diameter, inches	12
Water cylinder, diameter, inches	10
Stroke, inches	12
Piston rod (nickel steel), diameter, inches	2½
Pump rod (Tobin bronze), diameter, inches	2½
Valves, number in each pump	36
diamer of each, inches	2¾
area of opening through each, square inches	2.57

TRIAL DATA FOR FIRE AND BILGE PUMPS.

	Starboard.	Port.
Double strokes per minute	67.7	61.5
Indicated horsepower	15.2	13.6

Evaporator Feed Pumps.—There is one Blake, vertical, simplex, double-acting, piston, evaporator feed pump located in the evaporator room. This pump has a suction from the sea and from the circulating-water discharge from the distillers, for supplying warm feed to the evaporators; it discharges only into the evaporators.

DATA FOR ONE PUMP.

Steam cylinders, diameter, inches	4½
Water cylinders, diameter, inches	5
Stroke, inches	6
Piston rod (Tobin bronze), diameter, inches	1⅛
Valves (hard rubber), number for one pump	8
diameter, inches	2¼
area of opening through one, inches	2.57

The water pistons are packed with lignum vitae rings.

Distiller Fresh-Water Pump.—There is one Blake, vertical, simplex, fresh-water pump in the evaporator room, which takes the distilled water from the filter tank through a meter connection; this pump discharges into the ship's fresh-water tanks, forward and aft, to the main feed tanks, to the reserve-feed water tanks and to the tanks in the firemen's and servants' wash-rooms.

DATA FOR THIS PUMP.

Steam cylinder, diameter, inches	6
Water cylinder, diameter, inches	5
Stroke, inches	6
Piston rod, diameter, inches	$1\frac{5}{16}$
Pump rod (steel), diameter, inches	$1\frac{5}{16}$
Valves (hard rubber), number	8
diameter, inches	$2\frac{1}{4}$
area through one, inches, square	2.57

Engine-Room Oil Pumps.—In each upper engine room on the after bulkhead there is a Blake horizontal, duplex, piston, oil pump, with hand attachment, for filling the oil distributing tank. Each pump has a suction from the oil-storage tank and a discharge to the distributing tank, with an overflow pipe from the tank to the pump suction.

DATA FOR ONE PUMP.

Steam cylinder, diameter, inches	2
Oil cylinder, diameter, inches	$1\frac{1}{2}$
Stroke, inches	$2\frac{1}{4}$
Valves, number	8
diameter, inches	$1\frac{1}{8}$

Shaft Bilge Pump.—There is a bilge pump driven from a crank on the forward end of each main-engine shaft. The pump barrel is placed on the inner bottom directly over the crank-pit well, inboard of center line of engine. This pump may be used either on bilges or crank pits in its own compartment.

DATA FOR ONE PUMP.

Diameter of barrel, inches	8
Stroke, inches	8
Volume of discharge per stroke, cubic inches	402.12

FEED AND FILTER TANKS.

There is one feed and filter tank in each engine room, the capacity of each being about 5,525 gallons. The filter tank is placed over the feed tank proper, and is composed of three compartments, in which filtering material will be placed (probably loofa). The water from the main and aux-

iliary air pumps passes vertically through the three filters in succession and then flows into the feed tank.

Provision is made for cleaning out by removable plates over each filter.

There is a valve in the bottom of feed tank actuated by a float, to close when water goes below a certain level, to prevent the suction pipes and pumps from filling with air. Both tanks are connected through feed-suction pipe.

FEED-WATER HEATERS.

There is one feed-water heater in each engine room, of the high-pressure type; the feed water is discharged from the main feed pumps through the tubes of the heater, thence to the boilers. The auxiliary exhaust enters at the top of the shell and drains from the bottom to either the main or the auxiliary condenser. The shell is made of boiler steel, the tubes of Admiralty metal, the tube sheets of compostion and heads of cast steel.

DATA FOR ONE HEATER.

Diameter of shell, inside, feet and inches	2- 5½
Thickness of shell, inch	½
Length over heads, feet and inches	10- 4
Tubes, diameter outside, inch	⅝
length between tube sheets, feet and inches	7- 5
length, total ordered, feet and inches	7- 8½
thickness, No. 16 B.W.G., inch	.065
spacing between centers, inch	¾
number in each heater	683
Heating surface, both heaters, square feet	1,657.28

BOILERS.

There are twelve Babcock & Wilcox boilers, placed in six watertight compartments, with the axis of each boiler drum athwartships.

The arrangement of the boilers is most satisfactory. There is ample room at the sides and the back of each boiler for access to the cleaning and dusting doors and for cleaning and overhauling, taking out tubes, etc.; and there is a great deal of room over the boilers.

The boilers are of the design installed in the *West Virginia*, which has been described in the JOURNAL.

The boilers are each 25 sections in width, or "25-wide," the distance from center to center of sections being 7 inches.

The performance of the boilers on the trial was entirely satisfactory in every way. There was a slight leaking in the nipples in the front cross boxes or mud drums; this was due to the rigid manner in which the mud drums were held in between the corner boxes and the middle box, which permitted no expansion of the mud drum longitudinally. This has been remedied by cutting out one of the nipples from the inner end of one of the mud drums, leaving this drum free to expand in the direction of its length and leaving the middle box free to spring slightly to one side with the expansion of the other mud drum.

BOILER DATA.

Number of boilers	12
Total grate surface, square feet	1,097
Total heating surface, square feet	52,752
Ratio, grate surface to heating surface	48.1
Grate surface, one boiler, square feet	91.4
Heating surface, one boiler, square feet	4,396
Floor space, length one boiler, feet and inches	10- 1
width one boiler, feet and inches	14-10½
occupied by one boiler, square feet	149.98
all boilers, square feet	1,799.8
Length, over all, boiler rooms, feet	108
Width, over all, boiler rooms (each 23 feet), feet	46
Total area of boiler-room floors, square feet	4,968
Ratio boiler floor space to boiler-room floor space	.362
Inclination of tubes, degrees	15
Length of tubes, feet and inches	9- 2
Length of tubes between heaters, feet	9
Number of 4-inch circulating tubes in each element	1
2-inch tubes in each intermediate element	32
4-inch tubes in each intermediate element	1
2-inch tubes in each side element	25
4-inch tubes in each side element	2
2-inch tubes in one boiler	786
4-inch tubes in each boiler	53
Weight of water in one boiler at steaming level and temperature of 411 degrees, pounds	15,105
Weight of one boiler, complete, empty (mean of twelve), pounds	89,355
Ratio weight of dry boiler to wet boiler	.85

ASH HOISTS.

There is one ash-hoist engine for each pair of fire rooms, with clutches on shafting, suitably arranged for hoisting from either port or starboard fireroom. The engine is of the usual Williamson Navy type, and is located in the fireroom hatch on the main deck. Each may be operated from the main deck, where tracks and trolleys are provided for handling ash buckets, or from the boat deck for the purpose of coaling steam launches.

DATA FOR ONE ENGINE.

Number of cylinders	2
Diameter of cylinders, inches	4½
Stroke, inches	4½

BLOWERS.

There are twelve forced-draft blowers, two in each fireroom, of the Sturtevant inverted, vertical type, with opposite cranks. The blowers are suspended from the protective deck over the drums of the boilers. They take air from the fireroom ventilators and discharge directly into the firerooms through fan casings. The engines are enclosed to protect them from coal dust, etc., and are fitted with sight-feed oiling gear.

DATA FOR ONE BLOWER.

Steam cylinders, two, each, diameter, inches	6
Stroke inches	5
Piston rods, diameter, inches	1- $\frac{5}{16}$
Diameter of fan, inches	66
Area of induction nozzle, square inches	1,410.3
ejection nozzle (direct into fireroom), square inches	4,402.0475

DATA FROM TRIAL, ALL BLOWERS.

Revolutions per minute, mean	517
Air pressure, mean, inches	1.98
I.H.P.	444

DISTILLING APPARATUS.

There are three evaporators, of the usual Navy type, located over the after dynamo room on the berth deck. There are two vertical distillers located in the after fireroom hatch, on the

main deck level. The plant was designed to make 16,500 gallons of water per day and, during test, attained a rate of 17,040 gallons per day.

DATA FOR ONE EVAPORATOR.

Inside diameter shell (steel), feet and inches	5- 0
Thickness, one, inch	$\frac{5}{8}$
two, inch	$\frac{1}{2}$
Length, over heads and braces, feet and inches	5-11
Total length over all, feet and inches	6-10$\frac{1}{8}$
Heads (steel), thickness, inch	$\frac{1}{2}$
Tube sheets (comp)., thickness, inch	$\frac{3}{4}$
Tubes (brass) No. 12, B.W.G., number in each evaporator	96
outside diameter, inches	2
length between tube sheets, feet and inches	5- 5$\frac{11}{16}$
total, as ordered, feet and inches	5- 7$\frac{7}{16}$
heating surface, one evaporator, square feet	275
Capacity, one evaporator, gallons per day	5,500
Dry pipe (brass), diameter, inches	4
slots ($\frac{1}{8}$ inch$\times$130 degrees arc transverse) number	20
area ($\frac{1}{8}\times$4.537$\times$20 inches), square inches	11.344
Steam connections, to coil, diameter, inches	2$\frac{1}{2}$
Drain connections to trap, diameter, inches	1$\frac{1}{2}$

DATA FOR ONE DISTILLER.

Shell (cast iron), diameter inside, inches	18$\frac{1}{2}$
thickness, inch	$\frac{7}{16}$
length over all, feet and inches	6-11
Tubes (brass), No. 16, B.W.G., number	187
outside diameter, inch	$\frac{5}{8}$
length between tube sheets, feet and inches	5- 5
total ordered, feet and inches	5- 6$\frac{1}{2}$
Tube cooling surface, square feet	165.49
Capacity, gallons per day	8,250
Circulating water inlet and outlet, diameter, inches	4$\frac{1}{2}$
Steam inlet, diameter, inches	4$\frac{1}{2}$
Fresh-water outlet, diameter, inches	1$\frac{1}{2}$

GENERAL WORKSHOP.

The work shop is the most satisfactory in equipment that the writer has ever seen aboard a naval vessel. All the power tools are electrically driven, each independently of the others. The shop is very conveniently and comfortably located in the central part of the engine-room hatch. The following is the equipment of the shop, viz:

One 36-inch, back-geared engine lathe, 12 feet between centers.

One 14-inch, back-geared engine lathe, 5 feet between centers. (Both of these tools were made by the Lodge and Shipley Machine Tool Company of Cincinnati.)

One 16-inch crank-shaper, with 8 speeds, made by the Cincinnati Shaper Co.

One Universal Milling Machine, with a range of 22 inches by 6.5 inches by 18 inches made by the Cincinnati Milling Machine Co.

One 22.5-inch vertical drill, back geared.

One 12-inch sensitive drill, drilling holes up to ⅜ inch. (Both drilling machines were made by Jas. Clark, Jr., & Co., of Louisville.)

One emery-grinder, with two wheels, one for plane grinding, one for tools; wheels 2 inches by 12 inches—made by Jas. Clark, Jr., & Co., of Louisville.

Two vise benches (one vise on each).

Each of these tools has its own electric motor for driving it, and it is hoped that the noisy, dirty and troublesome shop engine has forever disappeared.

There is a compressed-air connection in the shop for pneumatic tools.

This equipment will probably be duplicated for the shop of the *Connecticut.*

REFRIGERATING APPARATUS.

The refrigerating plant is located on the starboard side of the berth deck, abreast the forward 12 inch turrets. Two 2-ton Allen dense-air ice machines are installed athwartships in a separate closed compartment, in which is also placed the ice-making tank (capacity 9 cakes).

The refrigerator is the next compartment abaft the ice-machine room, and is divided into five sections, viz: An entrance or air lock, and four refrigerating rooms, the whole being insulated with cork and covered on sides and top with sheet zinc and on bottom with sheet lead.

The air piping is so arranged that the cooling air for any circuit can be controlled from the ice-machine room. The following are the combinations in which the different circuits may be run, viz:

From machines to ice tank, to refrigerating rooms, to scuttle butts and return.

From machines to ice tank, to refrigerating rooms and return.

From machines to ice tank, to scuttle butts and return.

From machines to ice tank and return.

From machines to refrigerating rooms, to scuttle butts and return.

From machines to refrigerating rooms and return.

From machines to scuttle butts and return.

DATA FROM TEST.

After Machine.

Average temperature of atmosphere, degrees	73.6
refrigerating rooms, beginning, degrees	78.5
reduction of temperature in refrigerating rooms, degrees	28.3
temperature of water in scuttle butt, beginning, degrees	73
reduction in temperature of scuttle butts, degrees	9
Temperature of water in ice tank, beginning, degrees	46
Pounds of ice frozen in 4 hours 30 minutes	135
Average steam pressure, in pounds	129
pressure in compressor, pounds	260
expander, pounds	66.5
revolutions per minute	118

Forward Machine (4 hours 27 minutes).

Average temperature of atmosphere, beginning, degrees	78.6
refrigerating room, beginning, degrees	67.25
reduction in temperature of refrigerating room, beginning, degrees	21
temperature of water in scuttle butts, beginning, degrees	75.25
reduction in scuttle butts, degrees	13.5
Temperature in ice tank, degrees	41
Pounds of ice frozen in 4 hours 27 minutes	135
Average steam pressure, in pounds	140.5
pressure in compressor, pounds	260
expander, pounds	70
revolutions per minute	119

During each of the above tests, the cooling air took the following course: From machine to ice tank, then to refrigerating rooms, then to scuttle butts and returned to the machine, so that all the cooling surface in the system was in use.

DATA FOR ONE ICE MACHINE.

Cylinder, steam, diameter, inches	9¼
compressor, diameter, inches	7
expander, diameter, inches	5¼
circulating pump, diameter, inches	3½
primer pump, diameter, inches	2
Stroke of crank, inches	13
Piston rods, length over all, feet and inches	2– 8½
(3) diameter, inches	1⅝
Cooling coil, length, face to face of elbows, feet and inches	4– 2⅛
length, center to center engine coils, feet and inches.	3– 1⅝
complete spiral turns, number	18
diameter, mean, inches	12¾
tubing, outside diameter, inches	2
tubing, inside diameter, inches	1¼

HEATING SYSTEM.

There are 315 radiators installed in various parts of the ship for heating purposes. This heating system is divided into eleven circuits, in addition to which there is a separate one for the galley and bakery. Provisions are made for heating water in all the lavatories and bathrooms with steam from the heating system.

The radiators are drained into three traps located in the engine rooms, the galley and bakery circuit has a separate trap, also in the engine room, while the drains from the various water heaters are led directly into the auxiliary exhaust line.

STANDARDIZATION TRIAL.

The *Louisiana* left the dock at the shipyard, Newport News, Va., at 9:50 A. M., December 9, 1905, to proceed to the trial course off Rockland, Me., for her official standardization trial. There were on board 502 people, distributed as follows, viz:

Engineer's department	279
Steward's department	102
Representatives of inspectors (officers, draftsmen, etc.)	21
Government representatives	4
Shipyard officials and representatives	13
Foremen, special workmen, etc	78
Navigators (4) and ship's doctor	5

There were about 2,160 tons of coal in the ship, and her displacement was slightly over 16,000 tons.

An attempt was made to adjust compasses on December 9th, but bad weather prevented; so the ship was anchored inside the Virginia capes until the morning of the 10th, when the compass adjustment was made, and the ship proceeded northward. On the afternoon of the 10th, a series of informal, progressive speed trials was made, running at each speed for one-half hour under the natural draft, in a fresh N.W. breeze and a rough sea, with the following results, viz:

Run No.	*Mean revs.*	*I.H.P.*	*Notes.*
1	108.5	11,890	Comparatively shoal water, head sea and head wind.
2	96.5	8,314	
3	91	7,192	
4	85	6,260	

On December 11 a two-hour forced-draft run at various speeds was made by the contractors, with the following results (strong W.N.W. breeze and long sea), viz:

Time.	*Mean revs.*	*I.H.P.*	*Notes.*
8:15			
8:30	111.25	11,762	Deeper water than on 10th, with following sea and wind.
8:45	116.75	13,685	
9:15	121	15,516	
9:45	123.5	16,640	
10:15	125.7	17,881	

The data taken on these two days, under different conditions of sea and wind, are interesting for comparison. The points gotten on the run on December 11th, tally very closely with the five points obtained at high speeds on the official standardization trial. The points gotten on the runs on December 10th, lie on a curve considerably inside the curve obtained on official trial, the difference being marked and being due, probably, to the different weather conditions.

At 4 P. M. of December 12th, the *Louisiana* anchored in Rockland harbor.

The standardization trials were run on December 13. Twelve runs alternately North and South were made across the measured mile. As may be seen from the table on the speed-curve sheet (Plate I), the first five runs were made at full speed, the remaining seven at slower speeds, decreasing at the rate of about 4 revolutions per minute for each successive run. The weather was very favorable for the trials, being cloudy and overcast at beginning, fine and clear, at end, with a light N.W. breeze blowing and a smooth sea.

The following weight statement shows the conditions at the time of the standardization trials, viz:

Draught, forward, feet and inches	24- 0
aft, feet and inches	25- 3
mean, feet and inches	24- 7.5
Displacement, tons	16,070
Fresh water for use of engineers (in compartments B-84 to B-89 inclusive, and B-92 to B-98 inclusive), tons	369
Fresh water for sanitary and culinary purposes, tons	314
Salt water (ballast), tons	787
Fresh water in forward, after and gravity tanks, tons	39.5
Anchor and chain (anchor, 8 tons), tons	32
Miscellaneous weights, tons	12,892
Total, excluding coal, tons	14,434

The following table gives the results of the twelve runs, viz:

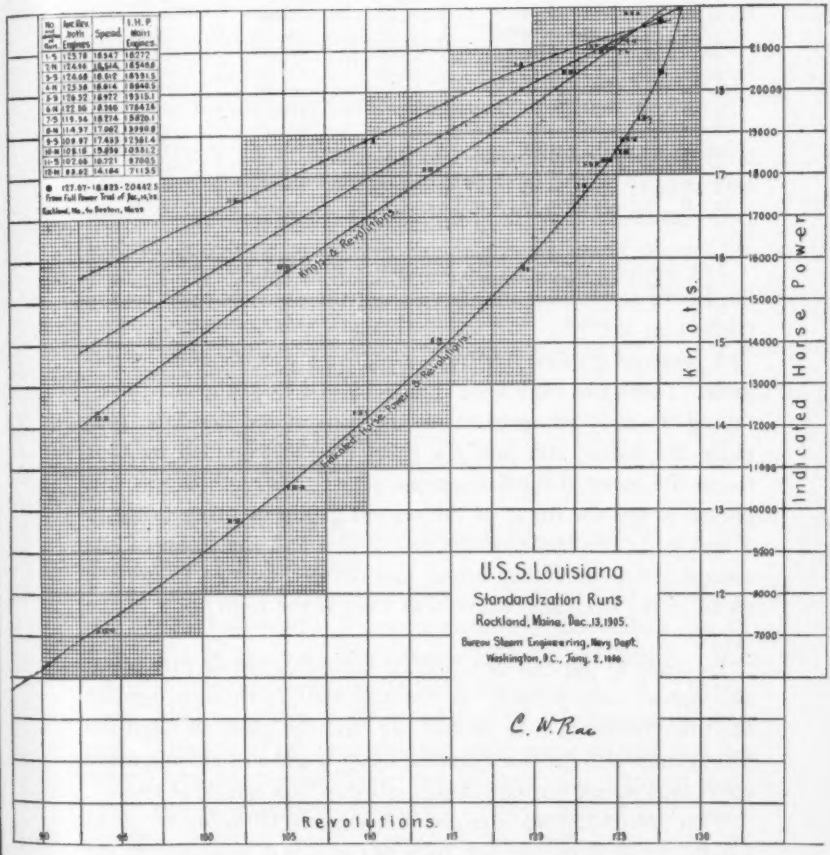

Knots.
Indicated Horse Power
Revolutions.
U.S.S. Louisiana
Standardization Runs
Rockland, Maine. Dec. 13, 1905.
Bureau Steam Engineering, Navy Dept.
Washington, D.C., Jany. 2, 1906
C. W. Rae

Plate I.

Number and direction of run.	Interval between middle of runs.	Average of elapsed times observed.	Mean revolutions per minute.	Speed, in knots.
	m. *s.*	*m.* *s.*		
1 S	... ...	3 14.1	123.78	18.547
2 N	18 52.28	3 14.45	124.96	18.514
3 S	18 31.31	3 14.47	124.68	18.512
4 N	19 42.16	3 13.40	125.38	18.614
5 S	19 00.33	3 10.25	126.32	18.922
6 N	18 41.42	3 17.3	122.56	18.246
7 S	18 37.85	3 17.0	119.34	18.274
8 N	20 44.6	3 31.0	114.37	17.062
9 S	19 41.95	3 26.5	109.97	17.433
10 N	21 26.75	3 47.0	105.18	15.859
11 S	19 54.85	3 35.3	102.66	16.721
12 N	23 29.35	4 13.8	93.62	14.184

The Speed Curve.—The following method was adopted by the Board in plotting the curve from the data obtained by the trials, viz:

A mean of the first five high-speed runs was obtained as follows: These five runs were divided into two groups of three; taking the first group of three, the sum of the first run, plus twice the second run plus the third run was divided by four to get the mean of the first group. A similar method was employed to get the mean of the second group of three (runs 3, 4 and 5). The mean of the first group and the mean of the second group were then added and divided by two to get the mean of the five runs, which was used as the high point of the curve.

In order to get the direction of the curve below this point, two curves were plotted, one through the North runs, against the tide (numbers 6, 8, 10 and 12), and the other through the South runs with the tide (numbers 5, 7, 9 and 11); then a mean curve was struck through points midway between these two.

This method gives a curve that crosses 18 knots at about 119.09 revolutions and the 19 knots at 129.9 revolutions; the mean point of five high-speed runs in 18.594 knots at 125.02 revolutions. As shown by the curve, the tide was variable, fluctuating from zero at beginning to .8 knot at end of runs.

At the end of the standardization runs, the draught was as follows:

Draught forward, feet and inches	24- 0½
aft, feet and inches	25- 0½
mean, feet and inches	24- 6½
Displacement, tons	16,006

The data obtained on the twelve runs over the measured mile are given in the accompanying tables (Tables I and II).

PRELIMINARY FOUR-HOUR OFFICIAL TRIAL.

At 7:35 A. M. of December 14, 1905, the *Louisiana* left the harbor of Rockland for her four-hours' official trial. The weather was fine and clear, with a light W.N.W. breeze and smooth, moderate sea. The following statement shows the draught and displacement on leaving and at end of trial, viz:

	Beginning.	End.
Draught forward, feet and inches	24- 6½	24- 3⅜
aft, feet and inches	24- 8½	24- 6⅜
mean, feet and inches	24- 7½	24- 4⅞
Displacement, tons	16,055	15,792
Feed water used during 4-hour run, tons		24.7
Culinary water used during 4-hour run, tons		3.6
Coal (bagged) used during 4-hour run, tons		75.2

The performance of the main engines and auxiliaries is given in the table which follows.

The working of the machinery, both main and auxiliary, was most satisfactory. The performance of the boilers was all that could be desired; they steamed freely and easily. The casing at sides and over tops of the boilers was cool at all times.

As usual on trials, a large amount of lubricating oil was used. Streams of oil were running from every oil pipe and the inward-turning cranks threw this oil over the working platforms and unfortunate bystanders. Oilers, armed with large syringes, stood at "ready," prepared to shoot at the most attractive, shining mark; one man was seen to squirt a copious stream on a cylinder-relief valve, which undoubtedly was large and dry. Of course, it is not expected that a contractor should take any chances of failure on a trial; but it seems that the use of oil should be regulated or restricted to a reasonable limit.

Table 1.—U. S. S. *LOUISIANA*.

Revolutions and Horsepower on Progressive Standardization Trials. Measured-Mile off Rockland, Maine, December 13, 1906.

Number and direction of run.	Time on course.	Revolutions of main engines.					Speed, in knots per hour.	Indicated horsepower.												
		To make 1 knot.		Per minute.				Starboard engine.					Port engine.					Total of both main engines.	Total all auxiliaries.	Total all machinery.	Main engines and auxiliaries in use for propulsion.
		Starboard.	Port.	Starboard.	Port.	Mean.		H.P. cylinder.	I.P. cylinder.	Forward L.P. cylinder.	After L.P. cylinder.	Total.	H.P. cylinder.	I.P. cylinder.	Forward L.P. cylinder.	After L.P. cylinder.	Total.				
	m. s.																				
1–S.	3–14.1	402.2	399.2	124.2	123.3	123.78	18.547	2,972	2,895.8	1,607.9	1,660.3	9,136	.....	.....	.....	.....	.....	.....	713.2	.....	.....
2–N.	3–14.45	403.3	403.2	124.9	124.9	124.96	18.514	2,990.2	2,979.7	1,617.7	1,758.4	9,346	2,891	2,998.5	1,669.8	1,643.5	9,202.8	18,548.8	817.4	19,366.2	19,302.1
3–S.	3–14.47	404.8	403 4	124.8	124.46	124.68	18.512	2,988	2,858.5	1,581.4	1,704.4	9,132.3	2,928.9	3,033.3	1,637.3	1,619.7	9,219.2	18,351.5	775.2	19,126 7	19,062.6
4–N.	3–13.4	405.5	405.2	125.4	125.3	125.38	18.614	3,050.2	3,023.7	1,623.5	1,720.6	9,418	2,973.8	3,107.4	1,701.5	1,639 8	9,422.5	18,840.5	752.5	19,593	19,528.9
5–S.	3–10.25	401.6	400.5	126.5	126.1	126.32	18.922	3,150.3	3,082.9	1,619 5	1,824.2	9,676.9	3,067.4	3,101.1	1,668.3	1,801.4	9,638.2	19,315.1	783.9	20,099	20,034.9
6–N.	3–17 3	402	403.2	122.38	122.7	122.56	18.246	2,940	2,788.1	1,394.7	1,635.7	8,758.5	2,895.6	2,893.7	1,528.2	1,571.4	8,888.9	17,647.4	654.3	18,301.7	18,237.6
7–S.	3–17	391.7	391 8	119.32	119.3	119.34	18.274	2,796.3	2,282.1	1,267.5	1,418.6	7,764.5	2,820.4	2,529.3	1,351.7	1,360.2	8,061.6	15,826.1	604.6	16,430.7	16,366.6
8–N.	3–31	400.3	404.4	113 79	114 96	114 37	17.062	2,800.5	2,073.9	1,024.6	1,120.7	7,019.7	2,615.3	2,131 8	1,091.8	1,132.2	6,971.1	13,990.8	555.2	14,546	14,481.9
9–S.	3–26.5	379.1	378.5	110.05	109.89	109.97	17.433	2,589.8	1,795.8	882.6	983.2	6,251.4	2,553 9	1,793.4	850. 4	912.3	6,110	12,361.4	449.4	12,810.8	12,746 7
10–N.	3–47	397.9	397.4	105.24	105.1	105.18	15.859	2,318.6	1,589.1	718.1	762.6	5,388.4	2,226.4	1,487	732.1	717.3	5,162.8	10,551.2	303.1	10,854.3	10,790.2
11–S.	3–35.3	370.1	367.2	103.05	102.26	102,66	16.721	2,122	1,446.9	717.7	775.7	5,062.3	2,005.5	1,354.5	654.6	683 4	4,698	9,760.3	293.8	10,054.1	9,990
12–N.	4–13.8	396.3	394.7	93.8	93 4	93.62	14 184	1,508.5	1,093.7	494.9	521 4	3,618.5	1,502	1,040	479 8	473.2	3,495	7,113.5	297.1	7,410.6	7,346.5

Table II.—U. S. S. *LOUISIANA*.

Data from Progressive Standardization Trials. Measured-mile, off Rockland, Maine, December 13, 1906.

Number of run.	Time at beginning of run, approximate.		Which engine.	Steam pressures.				Mean effective pressures.				Vacuum, inches.	Cut off in cylinders from beginning.			Temperatures.						Revolutions or double strokes.						Air pressure, inches of water.
	A. M.			In boilers above atmosphere.	In engine above atmosphere.	H.P. steam chest, gauge	In first receiver, absolute	In second receiver, absolute.	H.P. cylinder.	I.P. cylinder.	Forward L.P. cylinder.	After L.P. cylinder.	Referred to low pressure		H.P.	I.P.	L.P's.	Outside air.	Engine-room working platform.	Fireroom working level.	Injection.	Discharge.	Hotwell.	Feed.	Circulation pump.	Main feed pumps.	Main air pump.	Bilge pump.	Aux. condenser pump.	Fireroom blowers.	
1. S.	8 h. 15 m.	P.	S.	263.2	248	235	205	36	122	44	18.4	19	52.2	26.5	F	G		39	39	71.3	40	112	100	182	132	51	20	46	60	444	1.09
2. N.		S.	P.	263.2	250	240	200	32	...	...	...	...	...	26.6	F	G		39	39	71.3	48	110	100	179	138	53	15	45	35	444	1.09
3. S.	21.4 sec.	P.	S.	270.9	258	245	202	36	128	45	18.4	20	53.16	26.8	F	G		39	39	69.3	40	108	100	184	150	47	20	53	60	470	1.31
	8 h. 33 m.	P.	S.	268.9	255	242	202	32.7	118	43.3	18	18.7	52.36	26.6	F	G		39	39	69.3	46	108	101	186	141	63	15	105	33	470	1.31
4. N.	52.7 sec.	S.	P.	268.5	255	242	202	30	122	43.2	18	19.4	51.98	26.6	.76			39	39	68	46	110	94	183	144	41	20	52	60	466	1.30
	8 h. 53 m.	P.	S.	268.5	255	242	202	32.5	120	42	18.7	18.5	52.04	26.7	.76			38	39	68	46	106	99	183	148	65	14	54	32	466	1.30
5. S.	35.4 sec.	S.	P.	273.5	260	247	205	37.5	124	45.5	18.4	19.5	53.36	26.2	.76			38	39	67.3	40	112	96	182	144	40	20	48	60	466	1.35
	8 h. 13 m.	P.	S.	273.5	260	250	205	33	121	46.8	19.3	18.6	53.44	26.7	.76			38	39	67.3	47	105	94	174	134	62	16	60	34	466	1.35
6. N.	46.3 sec.	S.	P.	282.3	265	255	206	38	127	46	18.8	20.5	54.37	26.6	.76			38	39	68.5	40	113	94	184	144	44	20	74	60	458	1.36
	9 h. 1 m.	P.	S.	282.3	270	257	206	34	124	46.4	18.8	20.3	54.30	26.6	.73			38	39	67	46	109	97	180	152	64	17	72	34	458	1.36
7. S.	1.2 sec.	S.	P.	268.2	250	240	197	33	122.5	43	16.2	18	50.87	26.4	.73			38	38	67	46	109	90	178	150	48	20	74	60	394	1.09
	9 h. 21 m.	P.	S.	268.2	255	242	198	31.5	120.3	44.5	17.7	18.2	51.51	26.6	.73			38	38	67	46	112	96	180	148	59	17	66	32	394	1.09
8. N.	44.2 sec.	S.	P.	256.9	245	235	188	31	119.5	38.2	15.1	16.9	46.85	27.5	.67			38	38	68.2	47	98	85	154	138	38	20	60	35	381	.91
	9 h. 50 m.	P.	S.	256.9	245	235	187.5	27.5	120.5	40	16.2	16.2	48	26.8	.67			38	38	68	47	94	84	158	150	50	18	80	35	381	.91
	10 h. 10 m.	P.	S.	258	248	240	188	26	125.5	34.4	12.8	14	43.84	27.8	.56			38	38	67.5	47	94	76	148	140	38	20	60	36	359	.83
9. S.	39.8 sec.	S.	P.	259	245	243	187.5	24	116	33	13.5	14	43.10	27.5	.56			38	64	67.5	47	94	83	148	150	50	19	62	36	359	.83
	10 h. 30 m.	P.	S.	258	240	236	185	24.2	120	30.8	11.4	12.7	40.38	27.6	.5			41	62	70.8	46	82	72	118	156	42	20	60	36	259	.40
10. N.	24 sec.	P.	S.	256	245	242	180	20.4	118.5	38.8	11.8	11.8	39.51	27.6	.5			42	62	70.8	46	82	78	129	154	38	20	60	36	259	.40
	10 h. 51 m.	P.	S.	256.4	245	227	60	31.5	107.5	36.5	9.7	10.3	35.71	27.9	.5			42	62	70.5	46	80	68	118	148	37	20	58	32	394	.08
11. S.	40.5 sec.	P.	S.	256.4	260	225	55	17	108	36.7	9.9	9.7	34.69	27.5	.5			42	62	65	46	76	68	128	152	37	20	56	36	394	.08
	11 h. 11 m.	P.	S.	270.4	260	225	55	20.3	105	36.5	9.99	10.7	34.92	27.1	.5			42	62	69.7	46	86	69	142	154	38	20	66	36	381	.03
12. N.	41.1 sec.	P.	S.	270.4	262	228	54	16.7	110	25	9.1	9.5	32.65	27.1	.5			42	62	69.7	46	79	64	146	146	34	20	68	36	381	.03
	11 h. 33 m.	P.	S.	253.8	245	185	45	16.5	82	22	7.5	7.9	27.38	27.8	.5			44	75	65.8	40	76	72	180	112	22	20	72	60	359	.02
	51.2 sec.	P.		253.8	252	185	44	15.5	82	21	7.3	7.2	26.59	27.3	.5			44	70	65.8	46	76	65	146	134	39	23	48	35	...	.02

About 540 gallons of oil were used between 9:30 A. M. and 1:30 P. M.

The firing of the boilers was very well done. The fires were coaled and raked alternately by automatic electric signal, the signalling apparatus being one that was devised and made by the contractors. The furnaces were numbered 1, 3, 4, 2, as one faces the boiler. At intervals of about 40 seconds, the automatic-signal apparatus would ring a gong and light a cluster of lights on the outboard side of each fireroom, at which time one furnace in each boiler would be worked; beginning with furnace No. 1 on the left, the cycle of operations for one fireroom and its two boilers would be as follows, viz:

Forward Boiler.

1	3	4	2
Coal	Coal	Rake	Rake

Fire Room.

Rake	Rake	Coal	Coal
2	4	3	1

After Boiler.

After the four furnaces of one boiler had gone through one cycle, as shown above, the man at furnace No. 1, would start the next cycle by *raking* his fire, No. 2 coaling, No. 3 raking, No. 4 coaling; then No. 1 would begin again with *coaling*, etc. Each fire was coaled or raked every four minutes. The fires were carried at a thickness of from 6 to 7 inches; and the orderly, regular, uniform and efficient manner in which the firing was done was a pleasant change from the fire-at-will way, in which the boilers of the older ships were fired on their trials. The firemen on the *Louisiana's* trial were largely taken from the shipyard.

The coal used was in bags.

DATA OF FOUR HOURS' FULL-POWER FORCED-DRAFT PRELIMINARY TRIAL.

Steam Pressures. (*Mean of observations taken every quarter hour.*)

	Starboard.	*Port.*
Maximum steam pressure at boilers, pounds	276	
Mean steam pressure at boilers, pounds	270.4	
Maximum steam pressure at engines, pounds (H.P. chest)	247	245
Mean steam pressure in engines, pounds	242.1	236.5
H.P. steam chest, gauge, pounds	242.1	236.5
1st receiver (absolute), pounds	114	119
2d receiver (absolute), pounds	40.9	44.7
Vacuum in condensers, inches of mercury, mean	26.5	26.5
maximum	27	26

Revolutions or Double Strokes per Minute. (*Mean of observations taken every quarter hour.*)

	Starboard.	*Port.*
Maximum average revolutions per minute main engines for fifteen-minute period	128.57	128.40
Average revolutions, main engines, per minute, 4 hrs.,	127.66	127.69
Mean revolutions, both engines, per minute	127.68	
Pumps, main air	21.5	21.9
circulating	162.3	148.8
feed, d.s. per m., (forward and after.)	28.4 28.1	30.4 30.1
fire and bilge	66.7	61.5
auxiliary, air and circulating	30	0
Dynamo (1 100-kw.)	350	
Blower engines	517	
Speed of ship, in knots per hour	18.823	
Slip of propeller, in per cent. of its own speed, based on mean pitch	16.99	17.01
Air pressure in firerooms, in inches of water, mean	1.98	

Mean Effective Pressures in Cylinders, in pounds per square inch. (*Average of cards taken at half-hourly periods*).

	Starboard.	*Port.*
Main engines, H.P. cylinder	108.2	99.4
I.P. cylinder	49.6	57.3
F.L.P. cylinder	19.8	23.8
A.L.P. cylinder	22.4	23.4
Mean equivalent pressure, in pounds per square inch, referred to combined area of L.P. pistons	54.85	59.04

INDICATED HORSE POWER.

	Starboard.	*Port.*
Main engines, H.P. cylinder	2,707.1	2,485.5
I.P. cylinder	3,349.9	3,873.1

Main engines, F.L.P. cylinder	1,780.3	2,138.4
A.L.P. cylinder	2,006.9	2,101.3
Total	9,844.2	10,598.3
Collective H.P. of both main engines	20,442.5	

AUXILIARIES.

	Starboard.	*Port.*
Air pump, main	12.25	12.62
Circulating pump, main	48.5	38
Feed pumps, main	94.5	99.2
main	193.7	
Auxiliary condenser, air and circulating pumps	1.9	
Fire and bilge pumps	15.2	13.6
Forced-draft blowers	444	
Dynamo engines	128.2	
Total	908.97	
Collective I.H.P., main engines, air, circulating and feed pumps	20,747.57	
Collective, main and auxiliary engines in operation during trial	21,350.47	

COAL.

Kind and quality used on trial	New River hand picked.
Pounds per hour, main and auxiliary engines, during trial	42,112

DEDUCED DATA.

I.H.P. (total) per square foot of grate surface	19.462
main engines, air, circulating, and feed pumps, per square foot of grate surface	18.913
I.H.P., main engines, air, circulating, and feed pumps, per square foot of heating surface	.3933
Pounds of coal per I.H.P. per hour, collective, main engines, air, circulating and feed pumps	2.03
Pounds of coal per I.H.P. per hour, all machinery in operation	1.9727
square foot of grate surface per hour	.3839
heating surface per hour	.7979
Cooling surface (main condenser) square foot per I.H.P. (total)	.9718
Heating surface square feet per I.H.P. (total)	2.47

AIR COMPRESSOR.

In the port engine room there is a small electrically-driven air compressor, made by N. A. Christiansen, of Milwaukee. It can compress 50 cubic feet of free air to a pressure of 100 pounds per square inch per minute. From the compressor there is a 1½-inch copper main led along the port side of the middle-line bulkhead, with connections, at intervals, for pneumatic tools. There are two such connections in each engine room and in each boiler compartment. The connections in the boiler compartments may be used for tube-sweeper attachment; but the compressor is probably too small for use in sweeping tubes. There are two connections in the work shop and one in the evaporator room.

TEST OF A CURTIS MARINE TURBINE.

By Ensign W. G. Diman, U. S. N., Member.

By request of the Fore River Shipbuilding Company, a series of tests were made at the works of the E. W. Bliss Company, Brooklyn, N. Y., November 28 and 29, 1905, by a Board appointed by the Secretary of the Navy, of which Commander A. B. Canaga, U. S. N., was senior member, to determine the amount of steam used per brake horsepower by an experimental Curtis marine turbine at different revolutions per minute, and with steam varying in quality from saturated to about 50 degrees Fahrenheit, superheat. The tests were made at revolutions varying from 600 to 900, and with a brake horsepower of 1,000 to 1,250, which is only the conditions of the higher speeds. In naval practice the speed is a variable quantity, and only during short intervals is full power attained.

Figure 1 shows a plan of the arrangement of the experimental plant.

The turbine is of the Curtis marine pattern, built for experimental purposes, but embodying the essentials of such a turbine as is now building by the Fore River Shipbuilding Company for marine use.

There are two shells connected by piping as shown in figure 2. The first shell contains four "go-ahead" stages, and the second shell three "go-ahead" stages and one backing stage. The "go-ahead" stages each consist of three rotors, with two sets of fixed guide blades, intermediate and fixed to the casing. The backing stage consists of five rotors with four fixed guide blades. All rotors are 54 inches in diameter measured at middle of radial length of blades.

The blades of the rotors are continuous around the whole periphery and vary in pitch and length according to the de-

gree of expansion necessary at the position they occupy in sequence. The fixed guide blades vary in length of arc around the periphery, as well as in pitch and blade length in the same manner as the rotors.

Each stage is separated from the next stage by a diaphragm forming part of the casing. The shaft fits neatly through holes in these diaphragms with only sufficient clearance for water packing. The last "go-ahead" stage and the backing stage are in the exhaust shell; there was no steam connection to the latter stage, so no experiments on backing were made.

The course of the steam was as follows: From the boiler to the throttle it passed through a six-inch pipe, from the throttle it passed through a 4½-inch opening to the steam chest. From the steam chest the steam was admitted to the nozzles of the first stage, these nozzles being eight in number, each controlled by a valve. Only six of these nozzles were in use during the test. At the second stage there are sixteen nozzles, nine of which are controlled by valves; at the third stage there are sixteen nozzles, eight of which are controlled by valves; at the fourth stage, sixteen nozzles, eight controlled by valves; at the fifth stage, twenty-four nozzles, ten of which are controlled by valves; at the sixth stage, thirty nozzles, nine of which are controlled by valves; at the seventh stage, twenty-seven nozzles, twenty of which are controlled by ten valves. The pressures in the various stages are regulated by means of the valves at the nozzles. Each stage was drained into a small reservoir placed in a pit below the shell. As the pressures in the last four stages were below that of the atmosphere, each of these reservoirs were provided with two valves, one between it and the shell, and one between it and the drain pipe, so that by opening the upper valve the water collected in the reservoir, then closing the upper valve and opening the lower valve it flowed by gravity into a drain pipe leading to an adjacent pit containing a tank and scales for weighing.

The bearings of the turbine shaft all had forced lubrication, and on account of the small diameter of the shaft it was found necessary to locate a bearing inside each shell. This

would not be the case in a turbine built for actual use, where the size of the shaft would be increased. Some of the oil from these bearings entered the steam spaces and was run off and weighed with the drip water. The nozzles of the first stage were so made that the steam was expanded to increase its velocity, the nozzles of all other stages were of the non-expansion type.

The turbine shaft extended beyond the exhaust shell and was fitted with two water brakes for measuring the horsepower. The casings enclosing the discs of the water brake were accurately balanced on knife edges. The casings were fitted with pipes and funnels through which they received water from pipes fitted above them. Below the casings there was a large pipe and valve, so that by opening or closing the valve the amount of water in the casings was regulated and the number of revolutions of the turbine could be accurately adjusted.

Steam was supplied by two Stirling boilers, each having a grate surface of 67.5 square feet, and heating surface of 2,523 square feet. The draft was furnished by a blower working on the induction system. The pressure carried at the boiler was 265 pounds per gauge. The feed water was supplied by a duplex plunger pump of the Blake pattern, and was heated to about 200 degrees Fahrenheit before entering the boiler by a 600-horsepower Cochran feed heater using exhaust obtained from the auxiliaries.

There was a superheater of the Babcock & Wilcox type using oil as fuel. This superheater was entirely separate from the boiler, and was so arranged that the steam could be sent through it or by-passed at will.

The steam passed from the exhaust casing through a wide cast-iron nozzle to the top of a Wheeler condenser having a cooling surface of 2,700 square feet.

The air pump was a twin, vertical, single-acting, valveless, suction air pump of the Edwards patent. The desired vacuum of 28 inches was easily maintained, and was regulated by means of an air valve so as to keep it uniform at all times.

DESCRIPTION OF APPARATUS, INSTRUMENTS, ETC.

The condensed steam from the exhaust of the turbine was taken from the condenser by the air pump and discharged into a pipe leading to two cylindrical tanks, each tank supported by a pair of platform scales. The weight of the water was taken at intervals of six minutes on all tests, the water being instantaneously shifted at the end of each interval from one tank to the other.

The water from the casing drips was collected in the reservoir and was drained through a drain pipe to a pit where it was weighed at intervals, the weights and times being recorded.

The pressure in the bowl of nozzle of the first stage of the turbine was maintained at exactly 245 pounds per gauge by means of a throttle valve. A gauge was connected to the same bowl and the steam pipe leading to the gauge also led to a weighted gauge-testing apparatus, so that the readings of the gauge were verified at frequent intervals. The temperature of the steam was taken by a thermometer inserted just inside the bowl of the first nozzle at the same place that steam was taken by the gauge for record; readings of the temperature taken as nearly as possible every minute. On the test of November 28, 1905, from 2:36 P. M. to 3:24 P. M., steam direct from the boiler was used; on all other tests the steam was superheated.

The length of the arm from the brake casing was accurately measured and the distance from the center of the shaft to knife edges of support on the scales was found to be 63.023 inches or $\frac{33}{2\pi}$ feet. Readings of the scales were taken at intervals of about 1½ minutes.

The revolutions were taken by a counter so arranged that it could be thrown in or out of gear by means of a lever. These readings were checked by an attached tachometer.

The pressures were taken in the different stages, at the boilers and throttle, by gauges with pipes leading into the

steam spaces. The lower stages and condenser pressures were taken from mercurial columns. The readings of the barometer in the testing space during each test were observed and the vacuum in the exhaust shell was kept at a pressure of 2 inches of mercury (absolute). It will be noted that the readings recorded showed slightly more vacuum in the exhaust stage than in the condenser, this being due to the location of the nozzle for the condenser mercury column near the air cock used for regulating the vacuum.

METHOD OF MAKING THE TESTS.

The turbine was started and adjusted to the number of revolutions desired, the steam pressure being regulated by hand to 245 pounds per gauge in the bowl of first-stage nozzles, and the pressure in the exhaust shell or seventh stage regulated to 2 inches of mercury (absolute.) This regulation was obtained by men stationed and trained for the purpose. The revolutions were regulated by the valve controlling the outlet of water from the brakes.

The drips were weighed as frequently as possible, the times being recorded and the rate for the hour thus determined.

The tests varied in length of time according to the time necessary to catch and weigh the drips and check other data. All data recorded will be found in Tables 1 to 7.

The results of the test were worked up by the Special Instruction Class in Steam Engineering. The following formulæ for superheated steam was used, taken from an article by C. V. Kerr on "The Potential Efficiency of Prime Movers," see "Transactions of American Society Mechanical Engineers," Vol. XXV.

Available heat units $= H_1 - H_2$.

$H_1 = C_1 S_1 + h_1$.

$H_2 = T_2 \left(C_1 \log_\varepsilon \frac{T_s}{T_1} + \frac{r_1}{T_1} + Z_1 - Z_2\right) + q_2$.

$C = .48 + .00065p$ = specific heat of superheated steam.

p = gauge pressure.

S_1 = degrees of superheat.
h_1 = total heat of dry saturated steam at initial pressure.
T_s = absolute temperature of superheated steam.
T_1 = absolute temperature of dry saturated steam at initial pressure.
T_2 = absolute temperature of exhaust.
r_1 = latent heat of evaporation at initial pressure.
Z_1 = entropy of liquid at initial pressure.
Z_2 = entropy of liquid at exhaust pressure.
q_2 = heat of liquid.

For the test without superheat a Barrus throttling calorimeter was used and the moisture calculated from a calibration test of the same instrument made with dry saturated steam as given in following table:

CALORIMETER TEST.

Gauge reading, actual.	Gauge reading, corrected.	Upper thermom. reading, actual.	Upper thermom. reading corr.	Lower thermom. reading, actual.	Lower thermom. reading, corr.
pounds.	*pounds.*	*degrees F.*	*degrees F.*	*degrees F.*	*degrees F.*
250	248.7	402	408	314	315.5
255	254.5	404	410	315	316.6
260	260.	405	411	315	316.6
265	264.1	408	414	317	318.9
260	260.	406	412	314	315.5
255	254.5	403	409	314	315.5
250	248.7	402	408	314	315.5

Attention is particularly called to Table A, showing the average results of each test at different revolutions and the avererage steam pressure, vacuum, and water rate for each series of tests with revolutions varying from 600 to 900. The table shows how slight the variation in water rate was for all tests under the same conditions; *i. e.*, the same degree of superheat. This, however, should be expected, as the initial and final pressures were kept constant and the opening of the nozzle valves on the first stage was always the same. The slight variation in the water rate was probably more affected by the method of handling the drip than by any inherent condition of the turbine.

On the first test of November 28, (see Table 1), the drains on all stages, including the backing stage, were closed during the three runs, at 600, 700 and 900 revolutions, and the shells were drained only at the end of the series. From the great variation in the water rate it was concluded that the water condensed and gradually filled the shells on the first run at 700 revolutions, and acted afterwards as a water brake, and was carried over at intervals during the succeeding runs at 900 and 600 revolutions. This test was repeated on November 29th, from 1:46 P. M. to 3:37 P. M., with all drips closed except those on the reversing stage, which was kept drained, the results being recorded in Table 6.

After completing the tests with the turbine in motion, it was locked to the brake casing, and a test was made to determine the standing torque by letting the steam flow through under exactly the same conditions as in the previous tests with low superheat; the results of this test are recorded in Table 7. Unfortunately, the vacuum could not be maintained constant during this test, owing to a slight air leak that developed.

Lines of torque are laid down on the diagrams accompanying Tables 1-7 by taking the revolutions as abscissæ, and the torque as ordinates; in all cases such lines are evidently straight lines and cut the axis at the point shown by the standing torque test with remarkable accuracy.

It was assumed that the average water rate for each series of tests gave nearer the probable true rate than the recorded result of any individual test.

With the above facts in view, curves of water rate and efficiency were laid out (Plates 1–7 to accompany Tables 1–7) as follows: From the recorded torque on each individual test the torque line was drawn, and from this line the torque was taken at various revolutions and the brake horsepower calculated. The water rate was taken as the average per hour for each series, and the water rate per brake horsepower was thus obtained from any number of revolutions and the water-rate curve laid down.

The available energy of each pound of steam was calcu-

lated by the formulæ previously given, and the efficiency was obtained by dividing the actual energy delivered at the brake per pound of steam by the available energy as thus calculated. From these results efficiency curves were laid down.

As only one point could be obtained from Table 3, no curve is shown.

SUPPLEMENTARY TEST.

On January 19, 1906, the senior member of the Board conducted tests to determine the water rate at approximately cruising speeds. The results of these tests are shown in Tables 8-11, from which curves Plates 8-11, were constructed.

The first test consisted in running at practically maximum revolutions and horsepower. The results of this trial are shown in Table 8. The revolutions were 700-800, and the brake horsepower, 1,543.06 to 1,631.5. Eight sections were open in first nozzle.

In the second test the revolutions were reduced to 500 and then to 400, with brake horsepowers of 301.76 and 259.59, respectively. Two sections were open in the first nozzle.

In the third test the revolutions were again increased to 700 with two sections open in the first nozzle. The brake horsepower here was 353.33. From the above tests the efficiency and water-rate curves in Plates 8 to 11 were constructed. In all of the preceding runs the vacuum was held at about 28 inches in the last stage.

Finally, the turbine was run at 700 revolutions, with two sections open in the first nozzle, to determine the maximum possible vacuum. The vacuum obtained in the last stage was about 28.72. The data from this run are given in Table 12.

The barometer was 30.178.

Table A.—Averages of Tests.

Date, 1905.	Time.		Pressure in steam chest. Gauge.	Vacuum in exhaust stage. (Corrected) inches.	Degrees of superheat.	Revolutions per minute.	Brake pull, pounds.	Brake horsepower.	Water. Total per hour.	Water per B. H. P. per hour.	Drips.
	A. M.	P. M.									
November 28.	11:55	12:25	245	28	4.0	902	1,374.0	1,232.7	17,399	14.11	All Drips closed during test and shells drained at end of test.
November 28.	10:31	11:13	245	28	4.8	698	1,627.5	1,136.6	16,725	14.71	
November 28.	11:13	11:55	245	28	10.2	598	1,761.0	1,053.0	17,214	16.35	
Average	.....	.....	245	28	6.3	.....	.....	.....	17,113	.....	
November 28.	12:31	12:51	245—	28	8.5—	900.8	1,425.2	1,283.8	16,986	13.23	Drips all open.
November 28.	12:59	1:13	245—	28	4.1	593—	1,796.0	1,065.0	17,254	16.2—	
Average	.....	.....	245	28	6.3			.....	17,120	.....	
November 28.	2:30	3:24	250	28	Per cent of moisture = .801 of 1 per cent.	698.5	1,672.8	1,168.5	17,405	14 90	Drips all open.
November 28.	4:12	4:30	245	28	44 9	901	1,438.2	1,295.8	16,737	12 92	Drips all closed during test. Shells drained at end of test.
November 28.	4:48	5:00	245	28	46.6	695 8	8,677 8	1,167.5	16,795	14.38	
November 28.	4:30	4:48	245	28	50.5	598.9	1,798.5	1,077.1	16,750	15 55	
Average	.....	.....	245	28	47.3	.....	.....	.....	16,761	.....	
	A. M.	A. M.									
November 29.	11:09	11:40	245	28	8 5	893.8	1,424 6	1,273.4	17,112	13.4	Drips all open.
November 29.	10:45	11:09	245	28	8.3	798.2	1,541 5	1,130.4	17,054	13.9	
November 29.	10:15	10:45	245	28	11.5	699.9	1,667.3	1,167.0	17,132	14.7	
November 29.	11:40	12:10	245	28	9.5	600.6	1,800	1,081.1	17,198	15.9	
Average	.....	.....	245	28	9.5	.....	.....	.....	17,124	.....	
	P. M.	P. M.									
November 29.	2:37	3:01	245	28	10.2	898	1,385.8	1,244.6	17,458	14.03	All Drips closed except on reversing stage, other shells drained at end of test.
November 29.	2:11	2:37	245	28	8 1	802	1,492.0	1,196.6	17,290	14 45	
November 29.	1:46	2:11	245	28	7.8	696 5	1,668.8	1,162.3	17,074	14.69	
November 29.	3:01	3:37	245	28	8.1	597.1	1,767.4	1,055.4	17,339	16.43	
Average	.....	.....	245	28	8.5	.....	.....	.....	17,290	.....	
November 29.	3:53	4:11	245	27 39	8.5	0	2,533	.....	17,368—	.....	Drips open.

Table 1.—Test of Curtis Turbine for Experimental Purposes, November 28, 1905.

Number of stages, 7. Wheels in each stage, 3. Kind of blades, extruded. Drips all closed during run and shells emptied at end. Barometer 30.38 corrected.

Time.	Pressure record.									Quality of steam.				Power record.					Water record.				
		In shell stages.									Calorimeter. Thermom'r.				Revolutions.					Barrel weights.			
	Steam-chest gauge.	1st, gauge.	2d, gauge.	3d, column.	4th, column.	5th, column.	6th, column.	7th, column.	In condenser column.	Temperature at steam chest.	Upper.	Lower.	Degrees superheat.	Time, A. M.	Tachometer.	Counter.	Brake pull, pounds.	Brake horsepower.	Time, A. M.	1st, pounds.	2d, pounds.	Drips, pounds.	Total, pounds.
Conditions, 700 revolutions per minute.																							
10:37																							
10:39	245	.....	.....	.....	.....	.....	.....	.....	.....	402	.....	.....	.....	10:39	.....	702	1,630	.....	10:37	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	402	.....	.....	.....	.....	.....	693	1,629	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	402	.....	.....	.....	.....	.....	695	1,628	.....	10:43	.....	1,591		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	698	1,630	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	410	.....	.....	5 5	.....	.....	705	1,660	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	700	1,655	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	425	.....	.....	21.5	.....	.....	694	1,650	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	693	1,640	.....	.....	.....	.....		
10:47	245	66.8	22	2.90	11.35	20.80	26.22	28.38	28.25	418	.....	.....	13.5	.....	.....	690	1,637	.....	10:49	1,607	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	692	1,625	.....	.....	.....	.....		
10:54	245	67.2	22.4	3.02	10.95	20.70	26.20	28.38	28.28	412	.....	.....	7.5	10:58	.....	696	1,605	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	406	.....	.....	1.5	.....	.....	703	1,615	.....	.....	.....	.....		
10:57	245	66.7	22.2	2.93	10.92	20.65	26.15	28.38	28.28	406	.....	.....	1.5	11:02	.....	703	1,617	.....	10:55	.....	1,652		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	703	1,619	.....	.....	.....	.....		
10:58	245	.....	.....	.....	.....	.....	.....	.....	.....	406	.....	.....	1.5	.....	.....	698	1,619	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	701	1,620	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	403	.....	.....	.....	.....	.....	701	1,620	.....	11:01	1,672	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	703	1,615	.....	.....	.....	.....		
11:02	.....	.....	.....	.....	.....	.....	.....	.....	.....	402	.....	.....	.....	.....	.....	699	1,617	.....	.....	.....	.....		
11:03	245	67.2	22.5	3.10	10.78	20.65	26 12	28.38	28.28	407	.....	.....	2.5	.....	.....	698	1,619	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	414	.....	.....	9.5	.....	.....	.....	.....	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	414	.....	.....	9.5	.....	.....	.....	.....	.....	.....	.....	.....		
11:06	.....	.....	.....	.....	.....	.....	.....	.....	.....	414	.....	.....	9.5	.....	.....	.....	.....	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	410	.....	.....	5.5	.....	.....	.....	.....	.....	.....	.....	.....		
11:08	245	67.2	22.5	3.12	10.70	20.65	26.10	28.38	28.28	410	.....	.....	5.5	.....	.....	.....	.....	.....	11:07	.....	1,693		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	410	.....	.....	5.5	.....	.....	.....	.....	.....	.....	.....	.....		
11:13	245	67.2	22.5	3.10	10.70	20.63	26 10	28.38	28.28	413	.....	.....	8.5	.....	.....	.....	.....	.....	11:13	1709.5	.....		
Totals	245	402.3	134.1	18.17	65.40	124.08	156.89	.....	169.65	8186	.....	.....	108.5	.....	.....	13,967	32,550	.....	.....	4988.5	4,936		
Av'age	245	67.05	22.35	3.03	10.90	20.68	26 15	28.38	28.28	409.3	.....	.....	4.8	.....	.....	698.35	1627.5	.....	.....	99,245		184	16,725
								28. corrected.												1,654.1			

At 12:36 drip was 383 pounds = 184 pounds per hour.

Rate per hour = 16,541 + 184 = 16,725.

B.H.P. $= \frac{689.35 \times 1627.5}{1,000} = 1,136.61.$ Water per B.H.P. per hour $= \frac{16,725}{1,136.61} = 14.71$

11:22	245	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....		
11:24	.....	67.3	22.5	3.06	.....	.....	.....	.....	.....	414	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....		

11:22	245	.....	22.5	3.06	10.70	20.60	26.10	28.38	28.28	424	.....	.....	9.5	11:22	.....	600	1,757	.....	11:19	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
11:24	.....	67.3	.....	.....	.....	.....	.....	.....	.....	418	.....	.....	7.5	.....	.....	601	1,760	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
11:26	245	.....	.....	.....	.....	.....	.....	.....	.....	409	.....	.....	4.5	11:26	.....	601	1,761	.....	11:25	1,702	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
11:28	245	.....	.....	.....	.....	.....	.....	.....	.....	414	.....	.....	9.5	.....	.....	598	1,758	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
11:30	245	67.2	22.5	3.09	10.70	20.65	26.12	28.38	28.28	423	.....	.....	18.5	.....	.....	599	1,778	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
11:30	245	67.2	22.5	3.09	10.70	20.65	26.12	28.38	28.28	430	.....	.....	25.5	11:30	.....	597	1,775	.....	11:31	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
11:33	245	.....	.....	.....	.....	.....	.....	.....	.....	424	.....	.....	19.5	.....	.....	599	1,773	.....	.....	1,701	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
11:33	245	.....	.....	.....	.....	.....	.....	.....	.....	422	.....	.....	17.5	.....	.....	598	1,770	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
11:36	245	.....	.....	.....	.....	.....	.....	.....	.....	420	.....	.....	15.5	.....	.....	599	1,770	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
11:36	.....	67.4	22.5	3.10	10.70	20.65	26.12	28.38	28.25	416	.....	.....	11.5	11:33	.....	597	1,765	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
11:36	245	67.4	22.5	3.10	10.70	20.65	26.12	28.38	28.25	414	.....	.....	9.5	.....	.....	595	1,763	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
11:37	245	.....	.....	.....	.....	.....	.....	.....	.....	412	.....	.....	7.5	.....	.....	595	1,761	.....	11:37	1,702	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
11:39	245	.....	.....	.....	.....	.....	.....	.....	.....	413	.....	.....	8.5	.....	.....	596	1,765	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
11:43	245	67.6	22.6	3.10	10.70	20.60	26.12	28.38	28.25	410	.....	.....	5.5	11:43	.....	598	1,758	.....	11:43	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
11:43	.....	.....	.....	.....	.....	.....	.....	.....	.....	410	.....	.....	5.5	.....	.....	598	1,753	.....	.....	1,701	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
11:47	.....	67.7	22.8	3.12	10.70	20.62	26.12	28.38	28.25	410	.....	.....	5.5	.....	.....	597	1,752	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
11:47	245	.....	.....	.....	.....	.....	.....	.....	.....	413	.....	.....	8.5	11:47	.....	597	1,753	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
11:49	245	.....	.....	.....	.....	.....	.....	.....	.....	410	.....	.....	8.5	.....	.....	597	1,753	.....	11:49	1,702	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
11:49	.....	.....	.....	.....	.....	.....	.....	.....	.....	409	.....	.....	5.5	.....	.....	598	1,752	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
11:53	245	.....	.....	.....	.....	.....	.....	.....	.....	410	.....	.....	5.5	.....	.....	599	1,751	.....	11:55	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
Totals	.....	337.2	112.9	15.47	.....	103.12	.....	.....	141.31	8708	.....	.....	213.5	.....	.....	13156	36,741	.....	.....	5,106	1,701	.....	.....	.....	.....	.....	.....	.....	.....	1,701
Average	245	67.44	22.58	3.09	10.70	20.62	26.12	28.38	28.26	414.7	.....	.....	10.2	.....	.....	598	1,761	1,053	.....	.....	5,103	.....	.....	.....	.....	.....	.....	.....	.....	5,103
									28. corrected.												10,219									10,219
																					1,703									1,703

At 12:36 drip was 383 pounds, = 184 pounds per hour. Rate per hour = 17,030 + 184, = 17,214.

B.H.P. = $\frac{598 \times 1761}{1,000}$ = 1053. Water per B.H.P. per hour = $\frac{17,214}{1,053}$ = 16.35

12:00	.....	.....	.....	.....	.....	.....	.....	.....	.....	402	.....	.....	.....	12:00	.....	903	1,347	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
12:01	245	66.2	22.4	3.10	10.80	20.65	26.05	28.38	28.25	402	.....	.....	.....	.....	.....	906	1,350	.....	12:01	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
12:08	245	66.0	22.6	3.10	10.75	20.60	26.00	28.38	28.25	406	.....	.....	.....	.....	.....	903	1,357	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
12:12	.....	.....	.....	.....	.....	.....	.....	.....	.....	402	.....	.....	.....	.....	.....	902	1,367	.....	12:07	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	1717.5
12:12	.....	.....	.....	.....	.....	.....	.....	.....	.....	405	.....	.....	.....	12:12	.....	906	1,371	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
12:16	245	66.5	23.0	3.25	10.70	20.55	26 03	28.38	28.25	408	.....	.....	.....	.....	.....	905	1,377	.....	12:13	1,705	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	408	.....	.....	.....	.....	.....	897	1,378	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	412	.....	.....	.....	.....	.....	898	1,382	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	416	.....	.....	.....	.....	.....	899	1,382	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
12:19	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	904	1,384	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	413	.....	.....	.....	12:19	.....	904	1,381	.....	12:19	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	1,725
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	413	.....	.....	.....	.....	.....	899	1,384	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
12:24	245	66.5	22.8	3.18	10.75	20.55	26.00	28.38	28.25	410	.....	.....	.....	.....	.....	902	1,382	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
12:25	.....	.....	.....	.....	.....	.....	.....	.....	.....	408	.....	.....	.....	.....	.....	903	1,381	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
12:27	.....	.....	.....	.....	.....	.....	.....	.....	.....	414	.....	.....	.....	12:25	.....	902	1,381	.....	12:25	1718.5	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
12:27	.....	.....	.....	.....	.....	.....	.....	.....	.....	411	.....	.....	.....	.....	.....	903	1,380	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
Totals	980	2,652	90.8	12 63	43.00	82.35	104.08	113.50	113 0	6128	.....	.....	.....	.....	.....	14436	21,984	.....	.....	3443.5	3442.5	.....	.....	.....	.....	.....	.....	.....	.....	3442.5
Average	245	66.3	22.7	3.16	10.75	20.59	26.02	28.38	28.25	408 5	.....	.....	4.0	.....	.....	902.25	1374.0	1232.7	.....	6,886	.....	.....	.....	.....	.....	.....	.....	.....	.....	6,886
									28. corrected.											1,721.5										1,721.5

At 12:36 drip was 383 pounds, = 184 pounds per hour. Rate per hour = 17,215 x 184 = 17,699.

B.H.P. = $\frac{902.25 \times 1,374}{1,000}$ = 1,238.7. Water per B.H.P. = $\frac{17,699}{1,232.7}$ = 14.11.

Table 2.—Test of Curtis Turbine for Experimental Purposes, November 28, 1905. South Brooklyn, N. Y.

Number of stages, 7. Wheels in each stage, 3. Kind of blades, extruded. Drips all open during run. Barometer, 30.38 corrected.

Time.	Pressure record.									Quality of steam.				Power record.					Water record.				
	Steam-chest gauge.	In shell stages.							In condenser column.	Temperature at steam chest.	Calorimeter. Thermom'r.		Degrees superheat.	Time, A. M.	Revolutions.		Brake pull, pounds.	Brake horsepower.	Time, A. M.	Barrel weights.		Drips, pounds.	Total, pounds.
		1st, gauge.	2d, gauge.	3d, column.	4th, column.	5th, column.	6th, column.	7th, column.			Upper.	Lower.			Tachometer.	Counter.				1st, pounds.	2d, pounds.		
12:33	245	66.5	22.2	2.85	11.50	20.95	26.20	28.38	28.25	.....	.....	.....	.....	.....	.....	.....	.....	.....	12:37	1,592	.....		
12:34	.....	.....	.....	.....	.....	.....	.....	.....	.....	414	.....	.....	.....	12:34	.....	900	1,445	.....	12:43	.....	1,590		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	415	.....	.....	.....	.....	.....	890	1,445	.....	.....	.....	.....		
12:41	.....	.....	.....	.....	.....	.....	.....	.....	.....	412	.....	.....	.....	12:41	.....	898	1,435	.....	12:49	1,615	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	412	.....	.....	.....	.....	.....	906	1,421	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	415	.....	.....	.....	.....	.....	910	1,419	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	905	1,403	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	413	.....	.....	.....	.....	.....	903	1,420	.....	.....	.....	.....		
12:42	245	67.0	22.4	2.90	11.25	20 70	26.12	28.38	28.25	.....	.....	.....	.....	.....	.....	900	1,421	.....	.....	.....	.....		
12:50	245	67.0	22.6	2 97	11.15	20.60	26.10	28.38	28.25	.....	.....	.....	.....	12:51	.....	895	1,418	.....	.....	.....	.....		
12:51	.....	.....	.....	.....	.....	.....	.....	.....	.....	411	.....	.....	.....	12:53	.....	.....	.....	.....	.....	.....	.....		
Av'age	245	66.8	22.4	2.91	11.30	20.75	26.14	28.38	28.25	413	.....	.....	8.5	.....	.....	900.8	1425.2	1283.82	.....	4797		996 pounds per hour.	16,986
								28.00 corrected.															
														Water per H.P. = 13.23									
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	411	.....	.....	.....	12:59	.....	585	1,811	.....	.....	.....	.....		
12.59	245	.....	.....	.....	.....	.....	.....	.....	.....	414	.....	.....	.....	1:01	.....	586	1,810	.....	1:01	1649.5	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	414	.....	.....	.....	1:03	.....	588	1,810	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	414	.....	.....	.....	1:05	.....	588	1,810	.....	1:07	1650 5	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	406	.....	.....	.....	1:07	.....	590	1,803	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	408	.....	.....	.....	1:09	.....	599	1,785	.....	1:13	.....	1,638		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	403	.....	.....	.....	1:10	.....	600	1,776	.....	.....	4,938			
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	405	.....	.....	.....	1:11	.....	601	1,777	.....	.....	16,460			
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	1:13	.....	601	1,780	.....	.....	794			
1:00	245	67.8	22.6	3.07	18.80	20.45	26.10	28.38	28.25	408 5	.....	.....	4 0	.....	.....	.....	.....	.....	.....	17,254			
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....		
1:05	245	67.8	22.5	3.10	18.80	20.40	26 07	28.38	28.25	.....	.....	.....	.....	.....	.....	593	1,796	1,065	.....	.....	.....		
1:13	245	67.2	22.4	3 05	18.90	20.45	26.10	28 38	28.30	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	Drip=794 pounds per hour	
Av'age	245	67 6	22.5	3.07	18.83	20.43	26 09	28.38	28.27	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....		
								28.	corrected.														
																			Water=16.2 per H. P.				

Table 3.—Test of Curtis Turbine for Experimental Purposes, November 28, 1905.

Number of stages, 7. Wheels in each stage, 3. Kind of blades, extruded. Drips all open. Barometer 30.27 corrected.

Time.	Pressure record.										Quality of steam.				Power record.					Water record.				
	Steam-chest gauge.	In shell stages. 1st, gauge, column.	2d, gauge, column.	3d, gauge, column.	4th, gauge, column.	5th, gauge, column.	6th, gauge, column.	7th, gauge, column.	In condenser, gauge, column.	Temperature at steam chest.	Calorimeter. Thermom'r. Upper.	Lower.	Per cent. moisture.	Time.	Revolutions. Tachometer.	Counter.	Brake pull, pounds.	Brake horsepower.	Time.	Barrel weights. 1st, pounds.	2d, pounds.	Drips, pounds.	Total, pounds.
											aug.	*aug.*		2:36		702	1,649						
																702	1,670						
																692	1,667		2:42	1614.5			
																693	1,667						
																700	1,666						
																704	1,666		2:48		1629.0		
2:46	250	68.2	23.2	3.21	10.85	20.40	26 00	28.27	28.20		402	298	.928			705	1,663						
																705	1,676		2:41 to				
																701	1,668		2:56			226	
																701	1,667						
														2:52		705	1,658		2:54	1640.5			
																700	1 673		3:00		1635.0		
2:56	250	68.4	23.2	3.20	10.70	20 30	25.95	28 27	28.20		401	300				695	1,678		3:06	1633.0			
																695	1,678		2:56 to				
																698	1,673		3:11			302	
														3:09		697	1,675						
3:05	250	68 4	23.2	3.28	10.70	20.30	25.95	28.27	28.10		401	302				699	1,676		3:12		1638.0		
																696	1,683						
																697	1,681						
3:13														3:15		697	1,682		3:18	1637.5			
	250	68 8	23 4	3.30	10.65	20 30	25.95	28.27	28 20		402	300	.812			697	1,680						
																696	1,682						
																695	1,684						
											402	299	.870	3:20		692	1,683		3:11 to				
3:21	250	68.7	23.5	3.27	10.60	20.27	25.95	28.27	28.20		402	302	.696			696	1,675		3:26			271	
														3:24		701	1,672		3:24		1644.5		
																				1631.4	1636.6	799	
Av'ag.	250	68.5	23.3	3.25	10 70	20.31	25.96	28.27	28.2		401.7	300.2	.801			698.5	1672.8		Ave for 6	rage min	1634.0 lbs.	1065 lbs.	
								28. corrected.															

Average per hour = 16,400 (Tanks) + 1,065 (Drips) = 17,465.

$$B.H.P. = \frac{2\pi L\,N.P.}{33{,}000} \quad L = \frac{33}{2\pi} \quad \therefore B.H.P. = \frac{N\,P.}{1000} = 1168.5$$

Water per B.H.P. per hour = 14.895 lbs.

Table 4.—Test of Curtis Turbine for Experimental Purposes, November 28, 1905.

Number of stages, 7. Wheels in each stage, 3. Kind of blades, extruded. Drips all closed during test and shells drained at end of test.

Time.	Pressure record.									Quality of steam.				Power record.					Water record.				
	Steam-chest gauge.	In shell stages.							In condenser, gauge, column.	Temperature at steam chest.	Calorimeter. Thermom'r.			Time.	Revolutions.		Brake pull, pounds.	Brake horsepower.	Time.	Barrel weights.		Drips, pounds.	Total, pounds.
		1st, gauge.	2d, gauge.	3d, column.	4th, column.	5th, column	6th, column.	7th, column.			Upper.	Lower.	Degrees superheat.		Tachometer.	Counter.				1st, pounds.	2d, pounds.		
													deg.	P.M.									
4:12	245	66.5	22.7	2.95	10.70	20.50	26.00	28 27	28.15	442				4:12		893	1,456						
										448						899	1,447						
										444						904	1,452						
										445						906	1,438						
										450						895	1,435						
4:19	245	65.5	22.4	2 93	10.80	20.45	25.95	28.27	28.20	446						901	1,430						
										445						905	1,435						
										445						905	1,437						
										455						904	1,430						
										459				4:26		905	1,431						
4:24	245	65.5	22.4	2.90	10.75	20.40	25.90	28.27	28.18	459						897	1,435						
										455				4:28		898	1,433						
Total	735	197.5	67.5	8.78	32.25	61.35	77.85	84.81	84.53	5393						10812	17,259						
Av'ge	245	65.8	22.5	2.93	10.75	20.45	25.95	28.27	28.18	449.4			44.9			901.0	1438.2				465	16737	
								28. corrected.															
	B.H.P. $= \frac{N \times P}{1000} = \frac{901 \times 1438.2}{1000} = 1295\ 8.$ Water per B.H.P. per hour. $= \frac{16737}{1295.8} = 12.92.$																						
								28.27		448				4.54		689	1,690		4.54	1633.5			
								28.27		454				4.54		694	1,677		5.00		1632.5		
4:57	245	66 4	22.0	3.05	10.70	20.35	25.95	28.27	28.15	456				4.58		697	1,677						
								28.27		446				5.00		703	1,667						
Total										1804						2783	6,711			1633.5	1632.5		
																				3266.0			
Av'age	245							28.27		451			46.6			695.8	1677.8			1633.0		465.	
								28. corrected.															
	B.H.P. $= \frac{695\ 8 \times 1677.8}{1000} = 1167\ 5.$ Water per B.H.P. per hour $= \frac{16795}{1167.5} = 14.38$ lbs.																						
4:36	245	66.0	22.0	2.97	11.00	20.45	25.95	28.27		462				4:36		593	1,796		4:36		1635.5		
										460						597	1,790						
										452						601	1,795		4:42	1625.0			
										446						604	1,792						
										455						606	1,784						
										463						601	1,798						
4:43	245	66.1	22.0	3 05	11 00	20.50	26 00	28.27		456						595	1,800		4:48		1625.0		
										452						594	1,808						
4:46										448													
Total		132.1	44 0	6.02	22.0	40.95	51.95	56.54		4094						4791	14,372			1625 0	3260.5		
Av'age	245	66.1	22.0	3.01	11.0	20.48	25.98	28.27		454.9			50.5			598.9	1796.5			1628.50		465.	
								28. corrected.															
	B.H.P. $= \frac{N \times P}{1000} = \frac{598.9 \times 1796.5}{1000} = 1077.1.$ Water per B.H.P. per hour $= \frac{16750}{1077.1} = 15.55.$																						

Average per hour=16,372+465=16,737.

Average per hour=465.

Average per hour=465 lbs.

Average per hour 16795 lbs.

Average per hour=465.

Rate per hour= 16285+465=16750.

XU

B.H.P. = $\frac{N \times P}{1000} = \frac{598.9 \times 1795.5}{1000} = 1077.1$. Water per B.H.P. per hour = $\frac{1077.1}{\ldots}$

Table 5.—Test of Curtis Turbine for Experimental Purposes, November 29, 1905.

Number of stages, 7. Wheels in each stage, 3. Kind of blades, extruded. Drips open. Barometer, 29.75 corrected.

Time.	Pressure record.									Quality of steam.				Power record.					Water record.				
	Steam-chest gauge.	In shell stages.							In condenser column.	Temperature at steam chest.	Calorimeter. Thermom.		Degrees superheat, per cent. moisture.	Revolutions.			Brake pull, pounds.	Brake horsepower.	Time.	Barrel weights.		Drips, pounds.	Total, pounds.
		1st, gauge.	2d, gauge.	3d, column.	4th, column.	5th, column.	6th, column.	7th, column.			Upper.	Lower.		Time.	Tachometer.	Counter.				1st, pounds.	2d, pounds.		
													deg.										
11:18	.....	.....	.....	.....	.....	.....	.....	.....	.....	413	.....	.....	.....	.....	.....	885	1,439	.....	11:15	.....	1614.5	.....	
	.....	.....	.....	.....	.....	.....	.....	.....	.....	414	.....	.....	.....	.....	.....	884	1,438	.....	11:21	1,602	.....	.....	
11:23	245	67.5	23.2	3.21	10.45	20.00	25.45	27.75	27.65	413	.....	.....	.....	.....	.....	889	1,434	.....	.....	.....	.....	.....	
11:26	.....	.....	.....	.....	.....	.....	.....	.....	.....	411	.....	.....	.....	.....	.....	891	1,428	.....	11:25	.....	.....	244.5	
	.....	.....	.....	.....	.....	.....	.....	.....	.....	414	.....	.....	.....	.....	.....	898	1,422	.....	11:27	.....	1600.5	.....	
	.....	.....	.....	.....	.....	.....	.....	.....	.....	415	.....	.....	.....	.....	.....	897	1,422	.....	.....	.....	.....	.....	
11:32	245	67.7	23.0	3.17	10.45	20.00	25.45	27.75	27.65	414	.....	.....	.....	.....	.....	899	1,418	.....	.....	.....	.....	.....	
11:33	.....	.....	.....	.....	.....	.....	.....	.....	.....	413	.....	.....	.....	.....	.....	898	1,420	.....	11:33	1,606	.....	.....	
	.....	.....	.....	.....	.....	.....	.....	.....	.....	413	.....	.....	.....	.....	.....	896	1,419	.....	.....	.....	.....	.....	
	.....	.....	.....	.....	.....	.....	.....	.....	.....	413	.....	.....	.....	.....	.....	895	1,419	.....	.....	.....	.....	.....	
	.....	.....	.....	.....	.....	.....	.....	.....	.....	414	.....	.....	.....	.....	.....	890	1,420	.....	.....	.....	.....	.....	
11:38	.....	.....	.....	.....	.....	.....	.....	.....	.....	414	.....	.....	.....	.....	.....	903	1,416	.....	11:39	.....	1603.5	.....	
																			11:40			286	
Totals	490	135.2	46.2	6.38	20.90	40.00	50.90	55.50	55.30	49.61	.....	.....	.....	.....	.....	10,725	17,095	.....	.....	3,208	4818.5	530.5	
Av'age	245	67.6	23.1	3.19	10.45	20 00	25.45	27.75	27.65	413.4	.....	.....	8.9	.....	.....	893 8	1424.6	1273.4	.....	1,604	1606.2	265.3	171.12
								28. corrected.												1605.1			

Rate per hour = 16,051 + 1068 = 171.12.

B.H.P. = $\frac{893.8 \times 1424.6}{1,000} = 1273.4$ Water per B.H.P. = $\frac{17,112}{1,273.4} = 13.44$.

10:47	.....	.....	.....	.....	.....	.....	.....	.....	.....	414	.....	.....	.....	.....	.....	798	1,545	.....	.....	.....	.....	.....	
10:50	245	67 9	23.0	3.25	10.40	20.00	25.50	27.75	27.60	412	.....	.....	.....	.....	.....	793	1,542	.....	.....	.....	.....	.....	
10:53	.....	.....	.....	.....	.....	.....	.....	.....	.....	413	.....	.....	.....	.....	.....	796	1,544	.....	10:53	.....	1613	.....	

+ 1068 + 1068

Totals	490	135.2	46.2	6.38	20.90	40.00	50.90	55.50	55.30	49.01						16,725	17,093			3,208	4618.5	530.5	
Av'age	245	67.6	23.1	3.19	10.45	20 00	25.45	27.75	27.65	413.4			8.9			893 8	1424.6	1273.4		1,604	1606.2	265.3	171.12
								28.	corrected.											1605.1			

B.H.P. $= \frac{893.8 \times 1424\ 6}{1,000} = 1273.4$ Water per B.H.P. $= \frac{17,112}{1,273.4} = 13.44.$

800 revolutions per minute.—Rate per hour = 16,108 + 946 = 17,054.

10:47										414						798	1,545						
10:50	245	67 9	23.0	3.25	10.40	20.00	25.50	27.75	27.60	412						793	1,542						
10:51										413						796	1,544		10:51		1613		
										417						800	1,548						
										418						799	1,547						
										415						798	1,538		10:55			228	
10:56										410						798	1,542						
										411						800	1,534		10:57	1612.5			
										412						802	1,543						
										409						797	1.539						
										410						797	1,537		11:03		1607.5		
11:04	245	67.8	23.2	3.25	10.40	20 00	25.45	27.75	27.60	412						805	1,531						
11:09										413						795	1,547			16.10			
																			11:10			245	
Total	490	135.7	46.2	6.50	20.80	40.0	50.95	55.50	55.2	5,366						10,378	20,037			3222.5	3220.5	473	
Av'age	245	67.85	23.1	3.25	10.40	20 0	25.48	27.75	27.6	412.8			8.3			798 2	1541.5	1230.4		1611.3	1610.3	236.5	17,054
								28.	corrected.											1610 8			

B.H.P. $= \frac{798.2 \times 1541.5}{1,000} = 1230.4.$ Water per B.H.P. $= \frac{17,054}{1,230.4} = 139.$

700 revolutions per minute.—Rate per hour = 16,215 + 917.2 = 17,132.2.

10:15																695	1,664		10:15				
										412						698	1,665						
										420						700	1,671		10:21	1,612			
										424						702	1,670						
10:24	245	67.8	23.0	3.27	10.30	19.95	25.45	27.75	27.65	422						703	1,667		10:25			236	
										416						702	1,670						
10:27										414						702	1,669		10:27		1.621		
										414						700	1,667						
										417						700	1,669						
										415						701	1,676						
										412						699	1,667		10:33	1,627			
										413						703	1,667						
10:35										414						700	1,663						
10:36	245	67.8	23.0	3.29	10.30	19.95	25.45	27.75	27.60	413						699	1,662				1,623		
										410						696	1,660		10:39			222.5	
																			10:40	1624.5			
										422						698	1,672		10:45				
Total..	490	135.6	46.0	6.56	20.60	39 90	50.90	55 50	55 25	6,238						11,198	26,678			4863.5	3,244	458.5	
Av'age	245	67.8	23.0	3.28	10.30	19 95	25.45	27.75	27.63	416			11.5			699.9	1667.3	1167		1621.2	1,622	229.3	17132.2
								28.	corrected.														

B.H.P. $= \frac{2\pi \times L \times N \times P}{33,000}$, $L = \frac{33}{2\pi}$, B.H.P. $= \frac{N \times P}{1,000}$ B H.P. $= \frac{699\ 9 \times 1667.3}{1,000} = 1167.$ Water per B.H.P. $= \frac{17,132.2}{1,167} = 14.7$

600 revolutions per minute.—Rate per hour=16,490+708=17,198.

11:44										414						588	1,810		11:45	1641.5			
11:46	245	68.2	23.6	3.41	10 05	19.75	25.45	27.75	27.60	412						593	1,795						
										414						593	1,796						
										416						603	1,795						
11:50										414						604	1,787		11:51		1,655		
										412						605	1,789						
										411						610	1,788						
11:54	245	68.3	23.7	3.45	10.05	19.75	25.45	27.75	27.65	414						608	1,790		11:55			180	
										413						605	1,810		11:57	1643.5			
11:58										414						585	1,820						
12:03										416						595	1,800				1,650		
										415						605	1,800						
										414						607	1,795						
12:08										414						608	1,798		12:09	1655.			
										415						605	1,807		12:10			174	
										414						600	1,810						
12:12										414						599	1,813						
Av'age	245	68.25	23.65	3.43	10.05	19.75	25.45	27.75	27.63	414			9.5			600.6	1,800			1,649.0		177	17,198
								28.	corrected.														

B.H.P $= \frac{600.6 \times 1,800}{1,000} = 1,081.1.$ Water per B.H.P. per hour $= \frac{17,198}{1,081.1} = 15.91.$

Table 6.——Test of Curtis Turbine for Experimental Purposes, November 29, 1905.

Number of stages, 7. Wheels in each stage, 3. Kind of blades, extruded. All drips closed except on reversing stage shell; other shells drained at end of run.

Time.	Pressure record. Steam-chest gauge.	In shell stages. 1st, gauge. column.	2d, gauge. column.	3d, gauge. column.	4th, gauge. column.	5th, gauge. column.	6th, gauge. column.	7th, gauge. column.	In condenser, gauge, column.	Quality of steam. Temperature at steam chest.	Calorimeter. Thermom'r. Upper.	Lower.	Degrees superheat, per cent. moisture.	Power record. Time.	Revolutions. Tachometer.	Counter.	Brake pull, pounds.	Brake horsepower.	Water record. Time.	Barrel weights. 1st, pounds.	2d, pounds.	Drips, pounds.	Total, pounds.
P. M.				+ lbs.																			
2:42	245	68.5	24.0	3.65	9.80	19.65	25.20	27.61	27.50	417	.....	.....	12.5	2.42	893	.....	1,387		2:37	.....	.....		
2:44	245	.....	.....	.....	.....	.....	.....	.....	.....	414	.....	.....	9.5	.....	896	.....	1,385		2:43	1,694	.....		
.....	245	.....	.....	.....	.....	.....	.....	.....	.....	415	.....	.....	10.5	.....	896	.....	1,390		2:49	.....	1,688		
2:52	245	.....	.....	.....	.....	.....	.....	.....	.....	415	.....	.....	10.5	2.52	900	.....	1,390		2:55	1,695	.....		
.....	245	.....	.....	.....	.....	.....	.....	.....	.....	415	.....	.....	10.5	.....	900	.....	1,390		3:01	.....	1677.5		
.....	245	.....	.....	.....	.....	.....	.....	.....	.....	415	.....	.....	10.5	.....	898	.....	1,387		.....	.....	.....		
2:56	245	68.3	23.8	3.57	9.85	19.75	25.20	27.61	27.50	415	.....	.....	10.5	.....	903	.....	1,385		.....	.....	.....		
2:57	245	.....	.....	.....	.....	.....	.....	.....	.....	415	.....	.....	10.5	2.57	890	.....	1,383		.....	.....	.....		
.....	245	.....	.....	.....	.....	.....	.....	.....	.....	414	.....	.....	9.5	.....	897	.....	1,385		.....	.....	.....		
.....	245	.....	.....	.....	.....	.....	.....	.....	.....	414	.....	.....	9.5	.....	897	.....	1,385		.....	.....	.....		
3:00	245	.....	.....	.....	.....	.....	.....	.....	.....	414	.....	.....	9.5	3.00	904	.....	1,383		.....	.....	.....		
3:01	245	.....	.....	.....	.....	.....	.....	.....	.....	413	.....	.....	8.5	3.01	903	.....	1,380		.....	.....	.....		
Total..	.....	68.4	23.9	3.61	9.82	19.70	25.20	27.61 28 corrected	27.50	4,976	.....	.....	122.0	.....	10,777	.....	16,630		.....	3,389	3365.5		
Av'age	245	.....	.....	.....	.....	.....	.....	.....	.....	414.7	.....	.....	10.16	.....	898.1	.....	1385.8		.....	.....	6754.5		
																					16,886		

900 revolutions per min.

Average B.H.P. = 1244.6

Drip rate = 572.

Rate per hour = 17,458.
Rate B.H.P. per hour = 14.03.
Barometer, 29.61 corrected.

$$\text{B.H.P.} = \frac{2\pi \times L \times N \times P}{33{,}000} \qquad L = \frac{33}{2\pi} \qquad \therefore \text{B.H.P.} = \frac{N \times P}{1{,}000} = \frac{898.1 \times 1385.8}{1{,}000} = 1244.6$$

Number of stages, 7. Wheels in each stage, 3. Kind of blades, extruded. All drips closed except on reversing shell; other shells drained at end of run.

Time.	Pressure record. Steam-chest gauge.	In shell stages. 1st, gauge.	2d, gauge.	3d, column.	4th, column.	5th, column.	6th, column.	7th, column.	In condenser column.	Quality of steam. Temperature at steam chest.	Calorimeter. Thermom. Upper.	Lower.	Degrees superheat, per cent. moisture.	Power record. Time.	Revolutions. Tachometer.	Counter.	Brake pull, pounds.	Brake horsepower.	Water record. Time.	Barrel weights. 1st, pounds.	2d, pounds.	Drips, pounds.	Total, pounds.
				+ lbs.																			
2:21	245	68.2	23.8	3.57	9.80	19.70	25.25	27.61	27.5	413	.....	.....	8.5	2:17	790	.....	1,502		2:19	1,665	.....		
.....	245	.....	.....	.....	.....	.....	.....	.....	.....	412	.....	.....	7.5	.....	806	.....	1,490		2:25	.....	1671.5		
.....	245	.....	.....	.....	.....	.....	.....	.....	.....	413	.....	.....	8.5	.....	810	.....	1,487		2:31	1,670	.....		
.....	245	.....	.....	.....	.....	.....	.....	.....	.....	413	.....	.....	8.5	.....	803	.....	1,485		2:37	.....	1680.5		
.....	245	.....	.....	.....	.....	.....	.....	.....	.....	412	.....	.....	7.5	.....	802	.....	1,505		.....	.....	.....		
.....	245	.....	.....	.....	.....	.....	.....	.....	.....	413	.....	.....	8.5	.....	797	.....	1,490		.....	.....	.....		
.....	245	.....	.....	.....	.....	.....	.....	.....	.....	413	.....	.....	8.5	.....	802	.....	1,505		.....	.....	.....		
.....	245	.....	.....	.....	.....	.....	.....	.....	.....	413	.....	.....	8.5	.....	795	.....	1,490		.....	.....	.....		
2:30	245	68.6	23.8	3.65	9.85	19.70	25.25	27.61	27.5	412	.....	.....	7.5	.....	807	.....	1,479		.....	.....	.....		
.....	245	.....	.....	.....	.....	.....	.....	.....	.....	412	.....	.....	7.5	2:35	807	.....	1,507		.....	.....	.....		
.....	245	.....	.....	.....	.....	.....	.....	.....	.....	413	.....	.....	8.5	2:37	793	.....	1,470		.....	.....	.....		
														20 m.									
20 m.	245	68.4	23.8	3.61	9.83	19.70	25.25	27.61 28 corrected	27.5	412.6	.....		11) 89.5 8.12	11)	8,822 802	11)	16,411 1,492		.....	3,335	3,352 6,687		

800 revolutions per min.

Average B.H.P. = 1196.58.

Drip rate = 572.

Rate per hour = 17,190.
B.H.P. rate per hr. = 14.45.

$$\text{B.H.P.} = \frac{2\pi \times L \times N \times P}{33{,}000} \qquad L = \frac{33}{2\pi} \qquad \therefore \text{B.H.P.} = \frac{N \times P}{1{,}000} = \frac{802 \times 1492}{1{,}000} = 1{,}196\ 58$$

Rate = 16,718 per hour.

Number of stages, 7. Wheels in each stage, 3. Kind of blades, extruded. All drips closed except on reversing shell; other shells drained at end of run.

Time.	Pressure record. Steam-chest gauge.	In shell stages.							In condenser, column.	Quality of steam. Temperature at steam chest.	Calorimeter. Thermom.		Degrees superheat, per cent. moisture.	Power record. Time.	Revolutions				Water record. Time.	Barrel weights.			

Number of stages, 7. Wheels in each stage, 3. Kind of blades, extruded. All drips closed except on reversing shell; other shells drained at end of run.

Time.	Pressure record.									Quality of steam.				Power record.					Water record.				
	Steam-chest gauge.	In shell stages.							In condenser gauge, column.	Temperature at steam chest.	Calorimeter. Thermom.		Degrees superheat, per cent. moisture.	Time.	Revolutions		Brake pull, pounds.	Brake horsepower.	Time.	Barrel weights.		Drips, pounds.	Total, pounds.
		1st, gauge.	2d,	3d, column.	4th, column.	5th, column.	6th, column.	7th, column.			Upper.	Lower.			Tachometer.	Counter.				1st, pounds.	2d, pounds.		
				+ lbs.										P. M.									
1:55	245	68.6	23.8	3.00	10.10	19.75	25.30	27.61	27.5	412	.....	.....	7.5	1:52	693	.....	1,670		1:55	1,652	.....		
2:00	245	.....	.....	.....	.....	.....	.....	.....	.....	411	.....	.....	6.5	.....	710	.....	1,679		2:01	.....	1,638		
.....	245	.....	.....	.....	.....	.....	.....	.....	.....	411	.....	.....	6.5	.....	691	.....	1,682		2:07	1,650	.....		
2:01	245	.....	.....	.....	.....	.....	.....	.....	.....	412	.....	.....	7.5	2:01	690	.....	1,672		2:13	.....	1,661		
.....	245	.....	.....	.....	.....	.....	.....	.....	.....	411	.....	.....	6.5	.....	694	.....	1,669		.....	.....	.....		
2:04	245	68.7	23.7	3 50	10.05	19.72	25.27	27.61	27.5	411	.....	.....	6.5	.....	700	.....	1,669		.....	.....	.....		
.....	245	.....	.....	.....	.....	.....	.....	.....	.....	411	.....	.....	6.5	.....	697	.....	1,666		.....	.....	.....		
2:06	245	.....	.....	.....	.....	.....	.....	.....	.....	410	.....	.....	5.5	2:06	696	.....	1,670		.....	.....	.....		
.....	245	.....	.....	.....	.....	.....	.....	.....	.....	413	.....	.....	8.5	.....	696	.....	1,665		.....	.....	.....		
.....	245	.....	.....	.....	.....	.....	.....	.....	.....	416	.....	.....	11.5	.....	697	.....	1,667		.....	.....	.....		
.....	245	.....	.....	.....	.....	.....	.....	.....	.....	415	.....	.....	10.5	.....	700	.....	1,660		.....	.....	.....		
2:11	245	.....	.....	.....	.....	.....	.....	.....	.....	415	.....	.....	10 5	2:11	694	.....	1,657		.....	.....	.....		
.....	.....	68.65	23.75	3.25	10.07	19.73	25.29	27.61	27.5	4,948	.....	.....	94.0	19 m	8,358	.....	20,026		.....	3,302	3,299		
.....	.....	.....	.....	.....	.....	.....	.....	.....	28 corrected.		.....	.....	.....	.....	.....	.....	.....		.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	412.3	.....	.....	7.8	.....	696.5	.....	1668.8		.....	6,601	.....		

700 revolutions per min.

Drip rate = 572.

Rate per hour = 17,074.
B.H.P. rate per hour = 14 69.

Average B.H.P. = 1162.5.

$$\text{B.H P.} = \frac{2\pi \times L \times N \times P}{33{,}000} \qquad L = \frac{33}{2\pi} \qquad \text{B.H.P.} = \frac{N \times P}{1{,}000} = \frac{696.5 \times 1{,}668.8}{1{,}000} = 1{,}162.3$$

Rate per hour = 16,502.5.

Number of stages, 7. Wheels in each stage, 3. Kind of blades, extruded. All drips closed except on reversing shell; other shells drained at end of run.

Time.	Pressure record.									Quality of steam.				Power record.					Water record.				
	Steam-chest gauge.	In shell stages.							In condenser, gauge, column.	Temperature at steam chest.	Calorimeter. Thermom.		Degrees superheat, per cent. moisture	Time.	Revolutions.		Brake pull, pounds.	Brake horsepower.	Time.	Barrel weights.		Drips, pounds	Total, pounds.
		1st, gauge. column.	2d, gauge. column.	3d, gauge. column.	4th, gauge. column.	5th, gauge. column.	6th, gauge. column.	7th, gauge. column.			Upper.	Lower.			Tachometer.	Counter.				1st, pounds.	2d, pounds.		
				+ lbs.																			
	245	68.2	23.7	3.50	9.80	19.60	25.30	27.61	27.55	407	.....	.....	2.5	3:07	597	.....	1,791		3:07	1,689	.....		
	245	.....	.....	.....	.....	.....	.....	.....	.....	414	.....	.....	9 5	.....	600	.....	1,761		3:13	.....	1676.5		
	245	.....	.....	.....	.....	.....	.....	.....	.....	413	.....	.....	8.5	.....	608	.....	1,757		3:19	1675 5	.....		
	245	.....	.....	.....	.....	.....	.....	.....	.....	414	.....	.....	9.5	.....	594	.....	1,762		3:25	.....	1,676		
	245	.....	.....	.....	.....	.....	.....	.....	.....	413	.....	.....	8.5	.....	595	.....	1,773		3:31	1675.5	.....		
	245	.....	.....	.....	.....	.....	.....	.....	.....	412	.....	.....	7.5	3:24	590	.....	1,775		3:37	.....	1667.5		
	245	.....	.....	.....	.....	.....	.....	.....	.....	412	.....	.....	7.5	.....	588	.....	1,777		.....	.....	.....		
	245	.....	.....	.....	.....	.....	.....	.....	.....	412	.....	.....	7.5	.....	585	.....	1,772		.....	.....	.....		
	245	.....	.....	.....	.....	.....	.....	.....	.....	414	.....	.....	9 5	.....	605	.....	1,760		.....	.....	.....		
	245	.....	.....	.....	.....	.....	.....	.....	.....	414	.....	.....	9.5	.....	602	.....	1,760		.....	.....	.....		
	245	.....	.....	.....	.....	.....	.....	.....	.....	414	.....	.....	9.5	3:31	604	.....	1,754		.....	.....	.....		
	245	68.2	23.8	3.57	9.75	19.55	25.30	27.61	27.55	.....	.....	.....	.....	.....	.....	.....	.....		.....	.....	.....		
	.....	68.2	23.75	3.54	9.77	19.58	25.30	27.61	27.55	4,539	.....	.....	.....	.....	6,568	.....	1944.2		.....	5,040	5,020		
	.....	.....	.....	.....	.....	.....	.....	.....	28 corrected	.....	.....	.....	.....	.....	.....	.....	.....		.....	.....	.....		
	.....	.....	.....	.....	.....	.....	.....	.....	.....	412.6	.....	.....	8.1	.....	597.1	.....	1767.4		.....	10,060	.....		

600 revolutions per min.

3:07 to 3:31 P. M.

Drip rate = 572.

Rate per hour = 17,339.
B.H.P. rate per hour = 16.43.

Average per B.H.P. = 1055.35.

$$\text{B.H.P.} = \frac{2\pi \times L \times N \times P}{33{,}000} \qquad L = \frac{33}{2\pi} \qquad \text{B.H.P.} = \frac{N \times P}{1{,}000} = \frac{597.1 \times 1{,}767.4}{1{,}000} = 1{,}055.35.$$

Water rate, 1,676.7.

Table 7.—Test of Curtis Turbine for Experimental Purposes, November 29, 1905.

Number of stages, 7. Wheels in each stage, 3. Kind of blades, extruded. Drips open. Weighing torque.

Time.	Pressure record.									Quality of steam.				Power record.					Water record.				
	Steam-chest gauge.	In shell stages.							In condenser, gauge, column.	Temperature at steam chest.	Calorimeter Thermom'r.		Degrees superheat, per cent. moisture.	Time.	Revolutions.		Brake pull, pounds.	Brake horsepower.	Time.	Barrel weights.		Drips, pounds.	Total, pounds.
		1st, gauge.	2d, gauge.	3d, gauge.	4th, column.	5th, column.	6th, column.	7th, column.			Upper.	Lower.			Tachometer.	Counter.				1st, pounds.	2d, pounds.		
3:58	245	70 8	24.0	3.05	8 60	19.2	24.75	27 61	27 55	415	.....	.....	.....	3:58	.....	.....	2,525	.....	3:59	1727.5	.....	.....	.....
3:59	.....	.....	.....	.....	.....	.....	.....	27 50	.....	414	.....	.....	.....	3:59	.....	.....	2,534	.....	4:05	.....	1742.5	.....	.....
4:00	.....	.....	.....	.....	.....	.....	.....	27.40	.....	414	.....	.....	.....	4:00	.....	.....	2,534	.....	4:11	1740.5	.....	.....	17368.3
4:02	.....	.....	.....	.....	.....	.....	.....	27.25	.....	411	.....	.....	.....	4:02	.....	.....	2,536	.....		3+5210.5	.....	.....	26.0
4:03	.....	71.0	24.0	3.17	8.40	19.0	24.75	27.25	26.75	411	.....	.....	.....	4:03	.....	.....	2,538	.....		17368.3	.....	.....	17394.3
4:04	.....	.....	.....	.....	.....	.....	.....	27.00	.....	414	.....	.....	.....	4:04	.....	.....	2,535	.....	.....	.....	.....	.....	.....
4:05	.....	.....	.....	.....	.....	.....	.....	26.80	.....	416	.....	.....	.....	4:05	.....	.....	2,537	.....	.....	.....	.....	.....	.....
4:08	.....	.....	.....	.....	.....	.....	.....	26.55	.....	412	.....	.....	.....	4:08	.....	.....	2,533	.....	.....	.....	.....	.....	.....
4:10	.....	.....	.....	.....	.....	.....	.....	26.55	.....	413	.....	.....	.....	4:10	.....	.....	2.530	.....	.....	.....	.....	.....	.....
4:11	.....	71.0	24.2	3.25	8.10	19.0	24 65	26.55	26.35	413	.....	.....	.....	4:11	.....	.....	2,526	.....	.....	.....	.....	.....	.....
4:12	.....	.....	.....	.....	.....	.....	.....	26.55	.....	413	.....	.....	.....	4:12	.....	.....	2,530	.....	.....	.....	.....	.....	. .....
Av'age	245	71.3	24.1	3.16	8.36	19.1	24.71	27 00	26.88	413	.....	.....	8.5	.....	.....	.....	2,533	.....	.....	.....	.....	6½ lbs. in 15 min.=26 lbs. per hour.	.....

Table 8.——Test of Curtis Turbine for Experimental Purposes, January 19, 1906.

Number of stages, 7. Wheels in each stage, 3. Kind of blades, extruded.

Time.	Pressure record.									Quality of steam.				Power record.					Water record.				
		In shell stages.									Calorimeter. Thermom'r.				Revolutions.					Barrel weights.			
	Steam-chest gauge.	1st, gauge.	2d, gauge.	3d, gauge.	4th, column.	5th, column.	6th, column.	7th, column.	In condenser, column.	Temperature at steam chest.	Upper.	Lower.	Degrees superheat, per cent. moisture.	Time.	Tachometer.	Counter.	Brake pull, pounds.	Brake horsepower.	Time.	1st, pounds.	2d, pounds.	Drips, pounds.	Total, pounds.
														10.11									
.....	245	65.2	26.0	5.7	9.0	20.15	25.05	28.14	.....	.....	.....	.....	.....	10:15	.....	799	2,032	.....	10:17	2122.5	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	800	2,031	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	798	2,085	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	803	2,031	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	10:20	.....	803	2,032	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	805	2,036	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	803	2,036	.....	10:23	.....	2083.0		
.....	.....	65.0	25.4	5.7	9 0	20.15	25 05	28.14	.....	.....	.....	.....	.....	10:25	.....	800	2,037	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	800	2,037	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	801	2,038	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	803	2,039	.....	10:29	2095.5	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	10:30	.....	802	2,040	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	803	2,038	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	10:34	.....	804	2,039	.....	.....	.....	.....		
.....	.....	65.0	25.4	5.72	9.0	20.10	25.00	28.14	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	10:35	.....	2095.0	401.0	
Totals	.....	195.2	76.8	17.12	27.0	60.40	75.10	84.42	.....	.....	.....	.....	.....	.....	.....	11224	28,491	.....	.....	4218.0	4178.0	401'0	
Av'age	245	65.1	25.6	5.71	9.0	20.13	25.03	28.14	.....	.....	.....	.....	.....	.....	.....	801.7	2035.1	.....	.....	.....	.....	.....	

Water, lbs. per hour=21046.23 + 1533.0 (drips)=22579.23.

B.H.P. $= \frac{801.7 \times 2035.1}{1,000} = 1631.5$ Water rate $= \frac{22,579}{1631.5} = 13.84$ pounds per hour.

.....	245	65.2	25.5	5.75	8.85	20.05	25.00	28.18	.....	.....	.....	.....	.....	10:36	.....	701	2,201	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	699	2,202	.....	.....	.....	.....		
.....	.....	65.3	25.5	5.78	8.85	20.05	25 00	28.18	.....	.....	.....	.....	.....	.....	.....	700	2,206	.....	.....	.....	.....		
.....	.....	65 2	25 4	5.78	8.85	20.05	25 00	28.18	.....	.....	.....	.....	.....	.....	.....	700	2,204	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	699	2,301	.....	10:41	2110.0	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	700	2,202	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	701	2,203	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	700	2,202	.....	.....	.....	.....		
.....	.....	65.2	25.4	5.78	8.85	20.05	25.00	28.18	.....	.....	.....	.....	.....	10:48	.....	700	2,202	.....	10:47	.....	2120.5		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	10:49	.....	702	2,201	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	700	2,201	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	703	2,201	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	704	2,200	.....	10:53	2106	.....	365 5	
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	10:55	.....	703	2,198	.....	.....	.....	.....	.....	
.....	.....	65.2	25.2	5.78	8.80	20.05	25 00	28.18	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	
Totals	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	9,812	30,824	.....	.....	4216.0	2120.5	365.5	
Av'age	245	65 2	25.4	5.78	8.84	20 05	25.00	28.18	.....	.....	.....	.....	.....	.....	.....	700.85	2201.7	.....	.....	6702.0	18 minutes.		
																					22,340 per hour.		

Water lbs. per hour = 21,046.23 + 1,533 0 (drips) = 22,579.23.

B.H.P. $= \frac{2201.7 \times 700\ 85}{1,000} = 1543.06$ Water rate $= \frac{22,599}{1,543.0} = 14.633$ pounds per hour.

Table 9.——Test of Curtis Turbine for Experimental Purposes, January 19, 1906.

Number of stages, 7. Wheels in each stage, 3. Kind of blades, extruded. Two sections open in 1st nozzle.

Time.	Pressure record.									Quality of steam.				Power record.					Water record.					
	Steam-chest gauge.	In shell stages.							In condenser, column.	Temperature at steam chest.	Calorimeter. Thermom'r.		Degrees superheat, per cent. moisture.	Time.	Revolutions.		Brake pull, pounds.	Brake horsepower.	Time.	Barrel weights.		Drips, pounds.	Total, pounds.
		1st, gauge.	2d, gauge.	3d, column.	4th, column.	5th, column.	6th, column.	7th, column.			Upper.	Lower.			Tachometer.	Counter.				1st, pounds.	2d, pounds.		
.....	245	28.4	7.0	7.0	22 00	26.25	27 50	28 18	.....	414	.....	.....	.....	11:10	.....	501	601	.....	.....	.....	.....	Water per hour = 5661.9 + 111 2 (drips) = 5763 1/	
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	411	.....	.....	.....	.....	.....	501	603	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	410	.....	.....	.....	.....	.....	501	602	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	410	.....	.....	.....	.....	.....	501	600	.....	.....	.....	.....		
.....	.....	28.2	7 2	7.0	22.05	26.25	27.45	28.18	.....	411	.....	.....	.....	.....	.....	504	599	.....	11:16	571.5	.....		
.....	.....	28 2	7 2	7.0	22.05	26.25	27.45	28.18	.....	413	.....	.....	.....	11:20	.....	506	599	.....	.....	.....	.....		
.....	.....	28 2	7 2	7.0	22.05	26.25	27.45	28.18	.....	416	.....	.....	.....	.....	.....	506	600	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	415	.....	.....	.....	.....	.....	505	601	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	415	.....	.....	.....	.....	.....	503	600	.....	11:22	.....	572.0		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	415	.....	.....	.....	.....	.....	503	600	.....	.....	.....	.....		
.....	.....	28.2	7.2	7.0	22.05	26.20	27.45	28.18	.....	414	.....	.....	.....	.....	.....	503	600	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	413	.....	.....	.....	.....	.....	504	600	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	412	.....	.....	.....	.....	.....	506	599	.....	11:28	574.0	.....		
.....	.....	28.2	7 0	6.9	22.05	26.20	27.45	28.18	.....	412	.....	.....	.....	11:31	.....	506	599	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	412	.....	.....	.....	.....	.....	502	600	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	412	.....	.....	.....	.....	.....	500	599	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	412	.....	.....	.....	.....	.....	502	598	.....	11:34	.....	572.0	20 0	
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	412	.....	.....	.....	.....	.....	503	597	.....	.....	.....	.....	.....	
.....	.....	28 2	7 0	7.0	22 05	26.20	27 45	28 18	.....	412	.....	.....	.....	11:37	.....	502	599	.....	.....	.....	.....	.....	
Total	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	9559.0	11396.0	.....	.....	.....	.....	.....	
Av'age	245	28.23	7.1	7.0	22 04	26.23	27.46	28.18	.....	412.7	.....	.....	.....	.....	.....	503.1	599.8	.....	.....	.....	.....	Barometer, corrected, 30.178.	

B. H. P $= \frac{503.1 \times 599\ 8}{1000} = 301.76$. Water rate $= \frac{5763.1}{301.76} = 19.098$ lbs. per hour.

Table 10.—Test of Curtis Turbine for Experimental Purposes, January 19, 1906.

Number of stages, 7. Wheels in each stage, 3. Two sections open, 1st nozzle. Barometer corrected, 30.179.

Time.	Pressure record.									Quality of steam.				Power record.					Water record.				
	Steam-chest gauge.	In shell stages.							In condenser, column.	Temperature at steam chest.	Calorimeter. Thermom'r.		Degrees superheat, per cent. moisture.	Time.	Revolutions.		Brake pull, pounds.	Brake horsepower.	Time.	Barrel weights.		Drips, pounds.	Total, pounds.
		1st,	2d,	3d,	4th, column.	5th, column.	6th, column.	7th, column.			Upper.	Lower.			Tachometer.	Counter.				1st, pounds.	2d, pounds.		
.....	245	28.4	7.0	7.2	22.05	26.30	27.50	28.18	.....	413	.....	.....	.....	11:41	.....	404	645	.....	11:40	568.0	.....	Water per hour = $\frac{3651.9}{111.2} + 5763.1$	
.....	.....	28.4	7.0	7.2	22.05	26.30	27.50	28.18	.....	412	.....	.....	.....	.....	.....	403	645	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	411	.....	.....	.....	.....	.....	401	644	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	411	.....	.....	.....	.....	.....	402	644	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	411	.....	.....	.....	11:46	.....	401	644	.....	11:46	.....	574.0		
.....	.....	28.4	7.0	7.2	22.10	26.30	27.50	28.18	.....	412	.....	.....	.....	.....	.....	401	646	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	415	.....	.....	.....	.....	.....	401	646	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	415	.....	.....	.....	11:50	.....	401	645	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	414	.....	.....	.....	.....	.....	400	645	.....	11:52	574.0	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	412	.....	.....	.....	.....	.....	400	646	.....	.....	.....	.....		
.....	.....	28.4	7.0	7.2	22.10	26.30	27.50	28.18	.....	413	.....	.....	.....	.....	.....	410	642	.....	11:58	.....	568.5		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	415	.....	.....	.....	.....	.....	409	641	.....	.....	.....	.....		
.....	.....	28.4	7.0	7.1	22.10	26.25	27.50	28.18	.....	414	.....	.....	.....	12:02	.....	406	641	.....	12:04	571.0	.....		
Total.	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....		
Av'age	245	28.4	7.0	7.18	22.08	26.29	27.50	28.18	.....	412.9	.....	.....	.....	.....	.....	403.0	644.15	.....	.....	.....	.....		

B.H.P. = $\frac{403 \times 644.15}{1000} = 259.59$ Water rate = $\frac{5763.1}{259.59} = 22.201$ lbs. per hour.

Table 11.—Test of Curtis Turbine for Experimental Purposes, January 19, 1906.

Number of stages, 7. Wheels in each stage, 3. Kind of blades, extruded. Two sections open, 1st nozzle. Barometer corrected, 30.178.

Time.	Pressure record.									Quality of steam.				Power record.					Water record.				
	Steam-chest gauge.	In shell stages.							In condenser, gauge, column.	Temperature at steam chest.	Calorimeter. Thermom'r.		Degrees superheat, per cent. moisture.	Time.	Revolutions.		Brake pull, pounds.	Brake horsepower.	Time.	Barrel weights.		Drips, pounds.	Total, pounds.
		1st, gauge.	2d, gauge.	3d, column.	4th, column.	5th, column.	6th, column.	7th, column.			Upper.	Lower.			Tachometer.	Counter.				1st, pounds.	2d, pounds.		
.....	245	28.0	7 0	7.0	22 25	26.20	27.45	28.18	.....	414	.....	.....	.....	12:09	.....	695	515	.....	12:10	5.	557.5		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	414	.....	.....	.....	.....	.....	700	504	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	413	.....	.....	.....	.....	.....	693	510	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	415	.....	.....	.....	.....	.....	696	509	.....	.....	.....	.....		
.....	.....	28 0	7.0	7 2	22.25	26.25	27.45	28.18	.....	413	.....	.....	.....	.....	.....	697	507	.....	12:16	551.5	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	413	.....	.....	.....	.....	.....	697	508	.....	.....	.....	.....		
.....	.....	27.8	6.8	7.4	22.35	26.25	27 45	28 18	.....	413	.....	.....	.....	12:22	.....	698	507	.....	12:22	.....	548.5		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	414	.....	.....	.....	.....	.....	696	503	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	414	.....	.....	.....	.....	.....	700	506	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	413	.....	.....	.....	.....	.....	700	505	.....	.....	.....	.....		
.....	.....	27.5	6.8	7.4	22 35	26.25	27 40	28 18	.....	413	.....	.....	.....	.....	.....	699	503	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	413	.....	.....	.....	12:28	.....	700	501	.....	12:28	545 0	.....		
.....	.....	27.5	6.8	7.4	22.35	26.20	27 40	28.18	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....		
Total	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	8371.	.....	.....	.....	.....	.....	Water per hour = 3651.9 + 111.2 = 5763.1.	
Av'age	245	27.8	6.9	7.3	22.31	26.23	27.43	28.18	.....	413.5	.....	.....	.....	.....	.....	697 6	506.5	.....	.....	.....	.....		

$$\text{B.H.P.} = \frac{506.5 \times 697.6}{1000} = 353.33. \qquad \text{Water rate} = \frac{5763.1}{353.33} = 16.311 \text{ lbs. per hour.}$$

Table 12.—Test of "High Vacuum" Turbine for Experimental Purposes, January 19, 1906.

Number of stages, 7. Wheels in each stage, 3. Two sections, 1st nozzle open.

Time.	Pressure record.									Quality of steam.				Power record.					Water record.				
		In shell stages.									Calorimeter. Thermom'r.				Revolutions.					Barrel weights.			
	Steam-chest gauge.	1st, gauge.	2d, gauge.	3d, column.	4th, column.	5th, column.	6th, column.	7th, column.	In condenser, gauge, column.	Temperature at steam chest.	Upper.	Lower.	Degrees superheat, per cent. moisture.	Time.	Tachometer.	Counter.	Brake pull, pounds.	Brake horsepower.	Time.	1st, pounds.	2d, pounds.	Drips, pounds.	Total, pounds.
.....	245	27.5	6.8	7.5	22.45	26.35	27.90	28.72	.....	413	.....	.....	.....	12:30	.....	700	535	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	414	.....	.....	.....	.....	.....	699	510	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	413	.....	.....	.....	.....	.....	698	516	.....	12:.4	.....	538.5		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	413	.....	.....	.....	.....	.....	695	532	.....	.....	.....	.....		
.....	27.5	6.8	7.5	22.45	26.35	27.85	28.72	.....	.....	413	.....	.....	.....	.....	.....	700	525	.....	12:40	545 5	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	413	.....	.....	.....	.....	.....	703	527	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	413	.....	.....	.....	12:47	.....	704	532	.....	12:46	.....	543.5		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	413	.....	.....	.....	.....	.....	700	533	.....	.....	.....	.....		
.....	27.5	6.8	7 5	22.45	26.45	27.92	28.75	.....	.....	412	.....	.....	.....	.....	.....	699	534	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	412	.....	.....	.....	.....	.....	697	533	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	412	.....	.....	.....	.....	.....	696	526	.....	.....	.....	.....		
.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	412	.....	.....	.....	.....	.....	703	524	.....	.....	.....	.....		
.....	.....	27.5	6.8	7.5	22.45	26.45	27.90	28.72	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....		
Total.	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	30 minutes = 162.5	
Av'age	245	27.5	6.8	7.5	22.45	26.40	27.89	28.73	.....	412.8	.....	.....	.....	.....	.....	699.5	527.25	.....	.....	.....	.....		Water, pounds per hour = 5,651.9 + 111.2 = 57,631.1

$$\text{B.H.P.} = \frac{699.5 \times 527.25}{1000} = 368.812$$

$$\text{Water rate} = \frac{57631.1}{368\ 812} = 15.626 \text{ pounds per hour.}$$

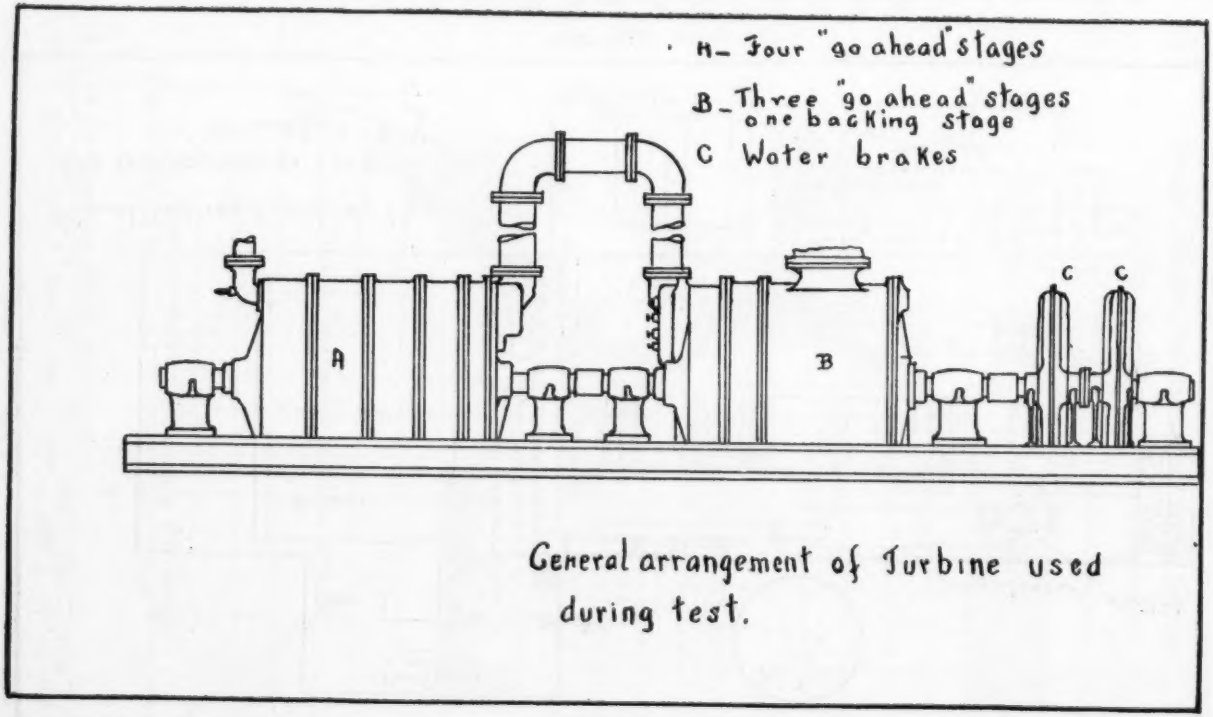

A- Four "go ahead" stages
B- Three "go ahead" stages
one backing stage
C Water brakes
A
B
C
C
General arrangement of Turbine used
during test.

Figure 1.

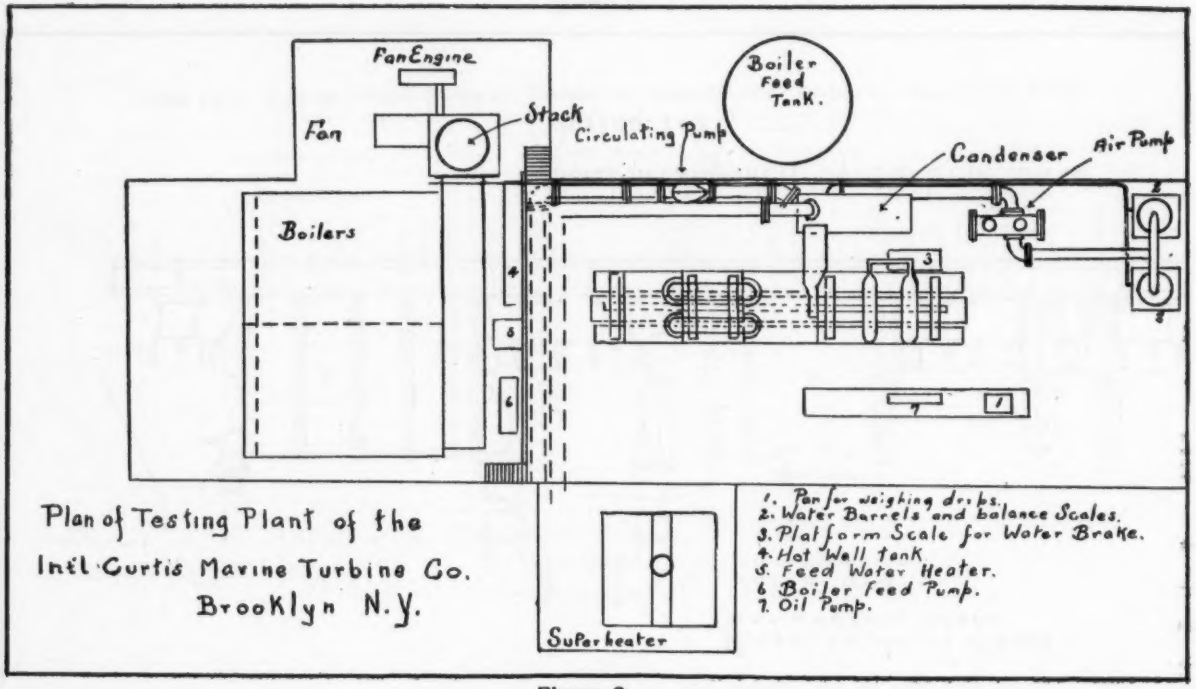
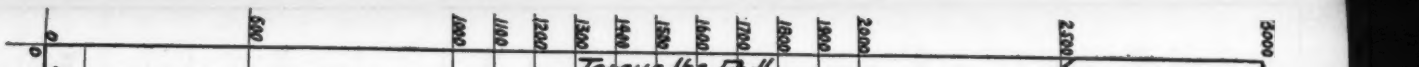

Fan Engine
Fan
Stack
Circulating Pump
Boiler Feed Tank.
Condenser
Air Pump
Boilers
Superheater
Plan of Testing Plant of the
Intl Curtis Marine Turbine Co.
Brooklyn N.Y.
1. Pan for weighing drips.
2. Water Barrels and balance Scales.
3. Platform Scale for Water Brake.
4. Hot Well tank
5. Feed Water Heater.
6 Boiler Feed Pump.
7. Oil Pump.

Figure 2.


0
500
1000
1100
1200
1300
1400
1500
1600
1700
1800
1900
2000
2100
3000

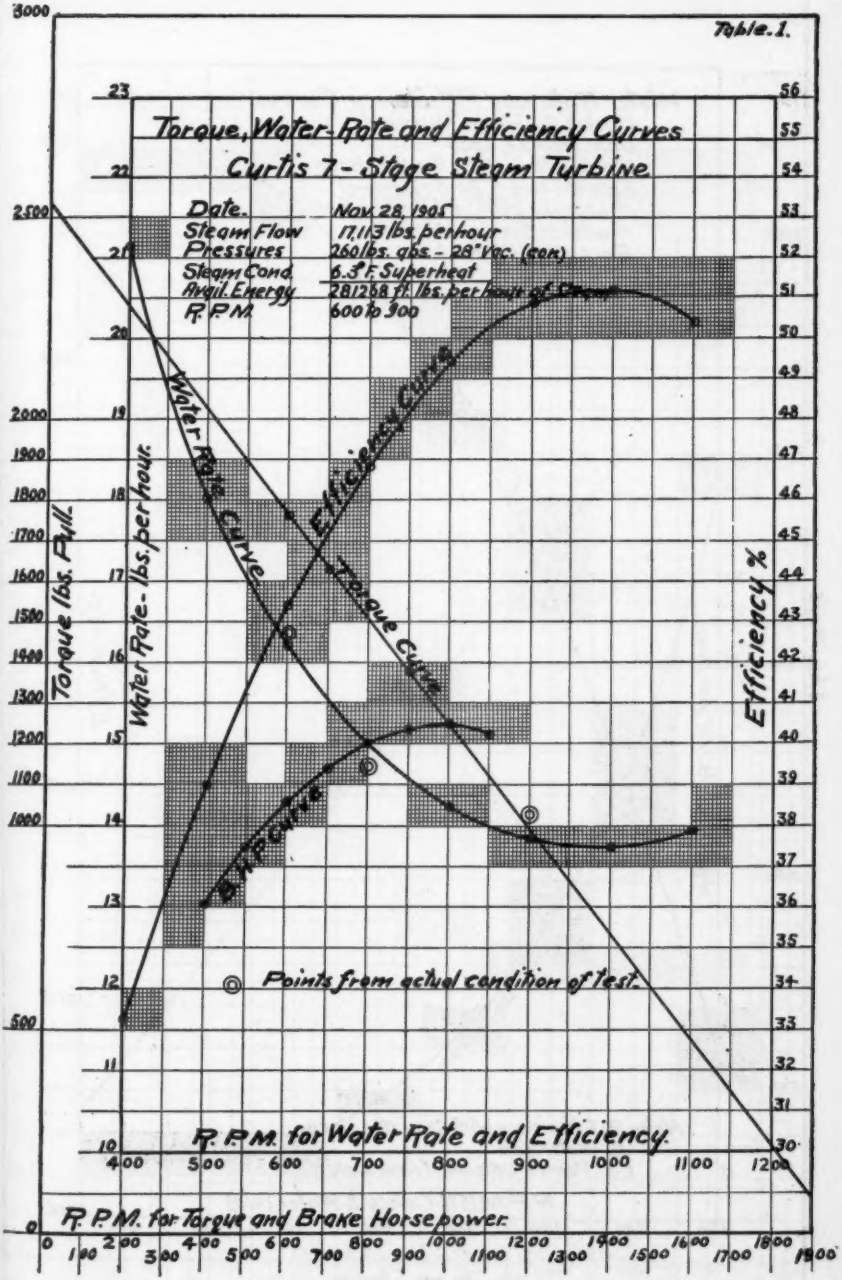


Table. 1.
Torque, Water-Rate and Efficiency Curves
Curtis 7-Stage Steam Turbine
Date. Nov. 28, 1905
Steam Flow 17,113 lbs. per hour
Pressures 260 lbs. gbs. - 28" Vac. (corr.)
Steam Cond. 6.3° F. Superheat
Avail. Energy 281268 ft. lbs. per hour per lb. Steam
R.P.M. 600 to 900
Torque Lbs. Pull.
Water Rate - lbs. per hour.
Efficiency %
Water Rate Curve.
Efficiency Curve.
Torque Curve.
B.H.P. Curve.
Points from actual condition of test.
R.P.M. for Water Rate and Efficiency.
R.P.M. for Torque and Brake Horsepower.

Plate 1. (Table 1).

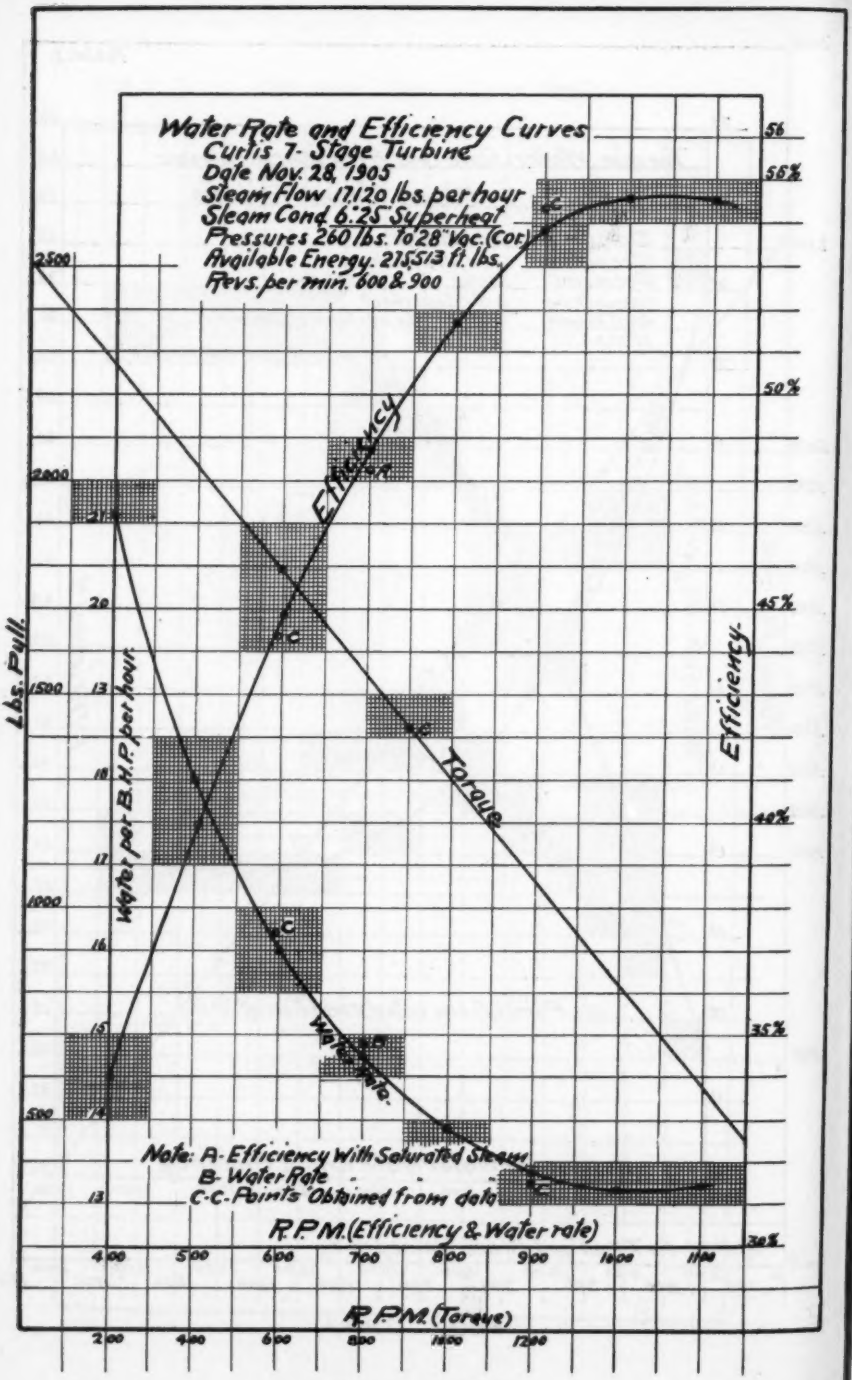


Plate 2. (Table 2).

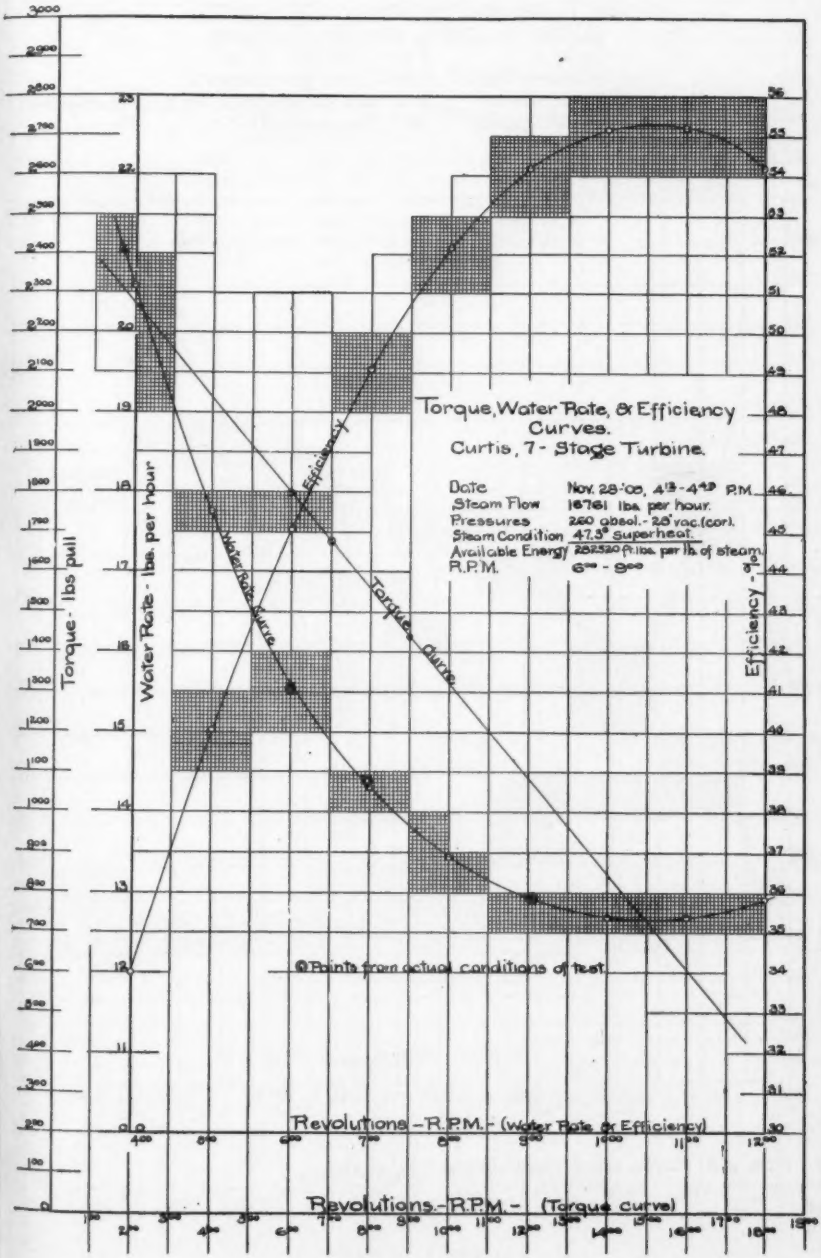
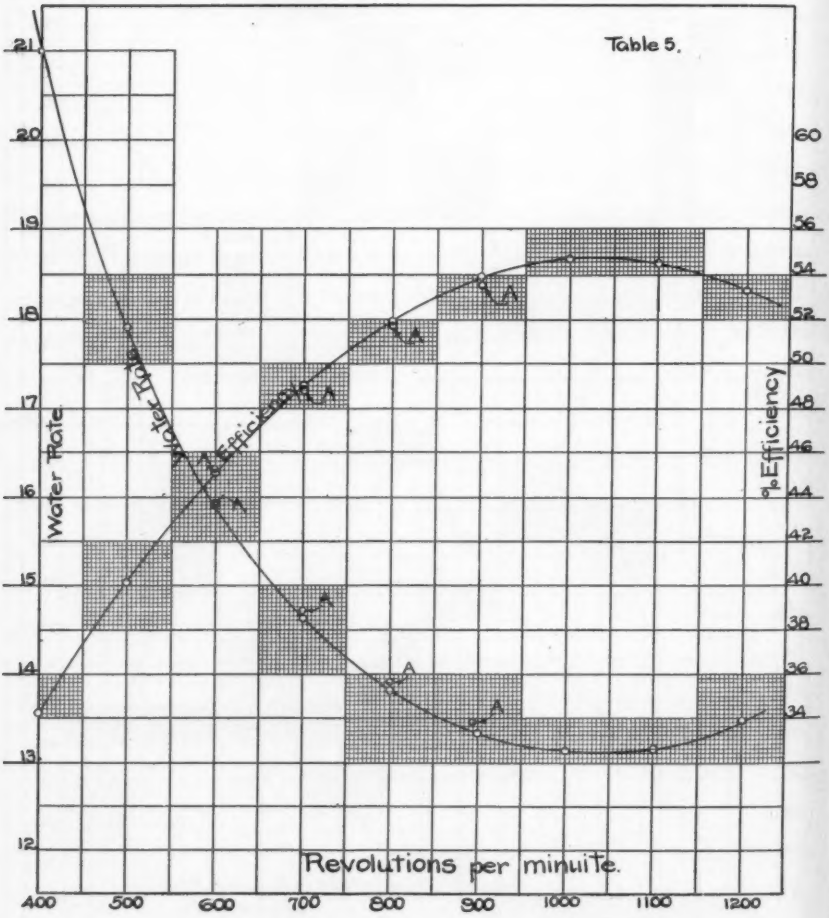

Torque, Water Rate, & Efficiency Curves.
Curtis, 7 - Stage Turbine.
Date Nov. 28-'05, 4¹³-4⁴² P.M.
Steam Flow 16761 lbs per hour.
Pressures 260 absol. - 26" vac. (cor).
Steam Condition 47.3° superheat.
Available Energy 282320 ft. lbs. per lb. of steam.
R.P.M. 6⁰⁰ - 9⁰⁰
Torque - lbs pull
Water Rate - lbs. per hour
Efficiency - %
Efficiency curve
Water Rate curve
Torque curve
@ Points from actual conditions of test.
Revolutions - R.P.M. - (Water Rate & Efficiency)
Revolutions - R.P.M. - (Torque curve)

Plate 4. (Table 4.)

Water Rate & Efficiency Curves

Curtis 7- Stag Steam Turbine.
Date - Nov. 29-'05. A.M.
Steam Flow 17100 lbs. per hour.
Pressures- 260 lbs. absolute to 28" vac. corrected.
Steam Cond. 9.5° Superheat.



Note-A-A- Points determined from trial data.

Plate 5. (Table 5.)

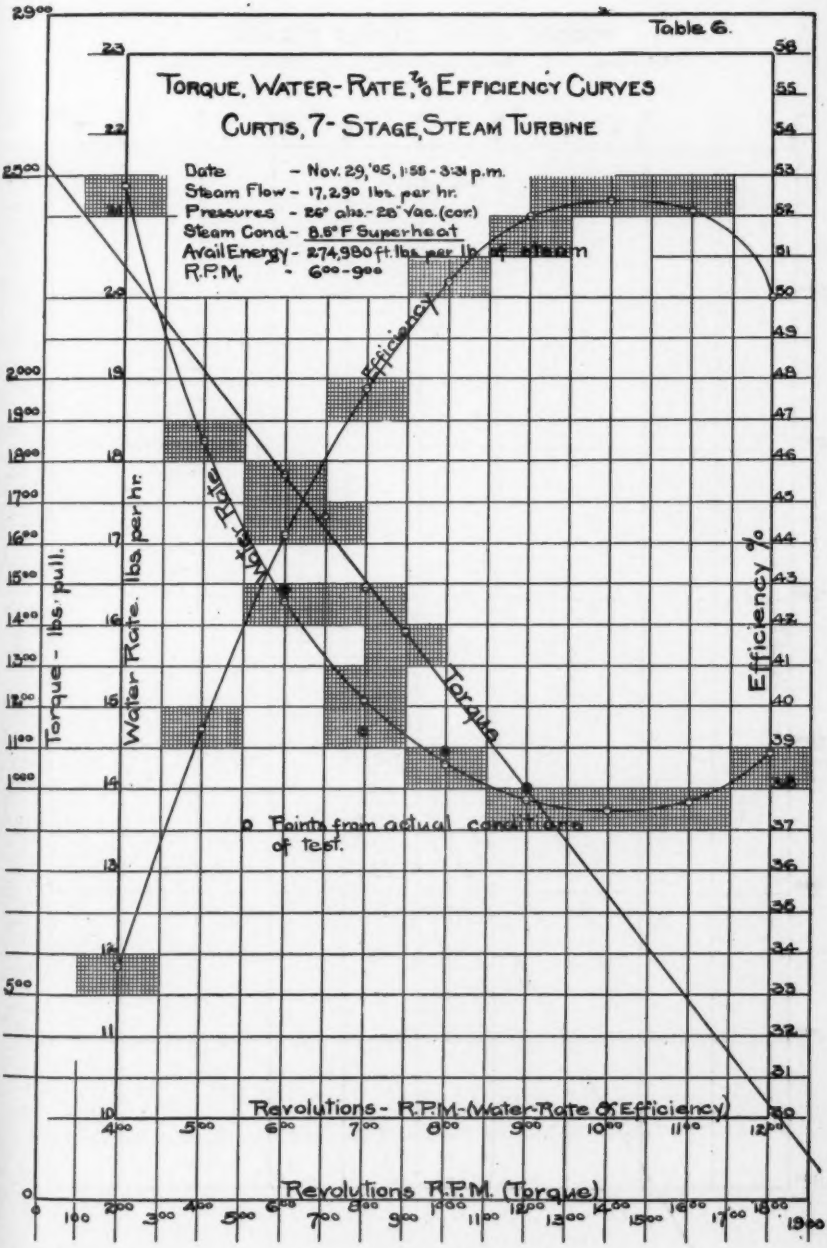


Table 6.
TORQUE, WATER-RATE, & EFFICIENCY CURVES
CURTIS, 7-STAGE, STEAM TURBINE
Date – Nov. 29, '05, 1:55 - 3:31 p.m.
Steam Flow – 17,290 lbs per hr.
Pressures – 26" abs. - 28" Vac. (cor.)
Steam Cond. – 8.6° F Superheat
Avail Energy – 274,980 ft. lbs per lb. of steam
R.P.M. – 600 - 900
Efficiency
Water-Rate
Torque
Torque - lbs. pull.
Water Rate. lbs per hr.
Efficiency %
Points from actual conditions of test.
Revolutions - R.P.M. - (Water-Rate & Efficiency)
Revolutions R.P.M. (Torque)

Plate 6. (Table 6.)

TORQUE CURVE, CURTIS TURBINE.

Drips open; Steam, superheated; Date, Nov. 29, 1905.

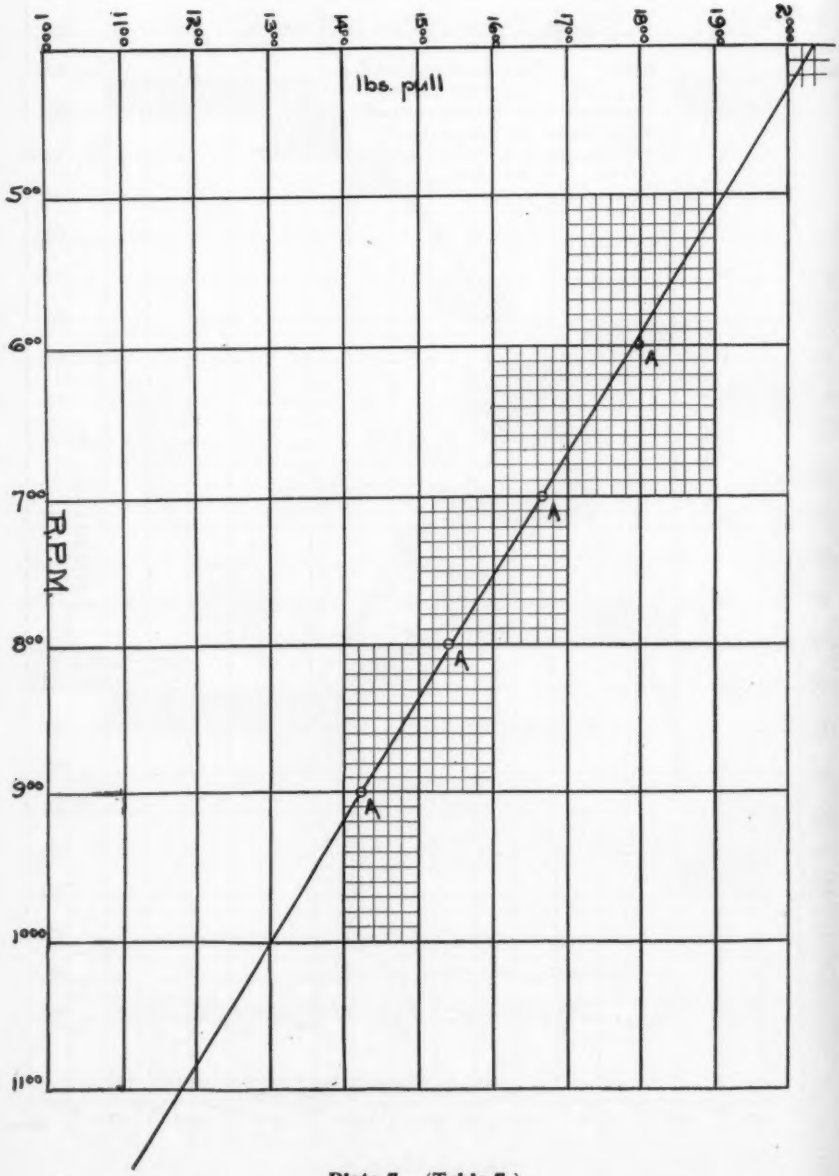


Plate 7. (Table 7.)

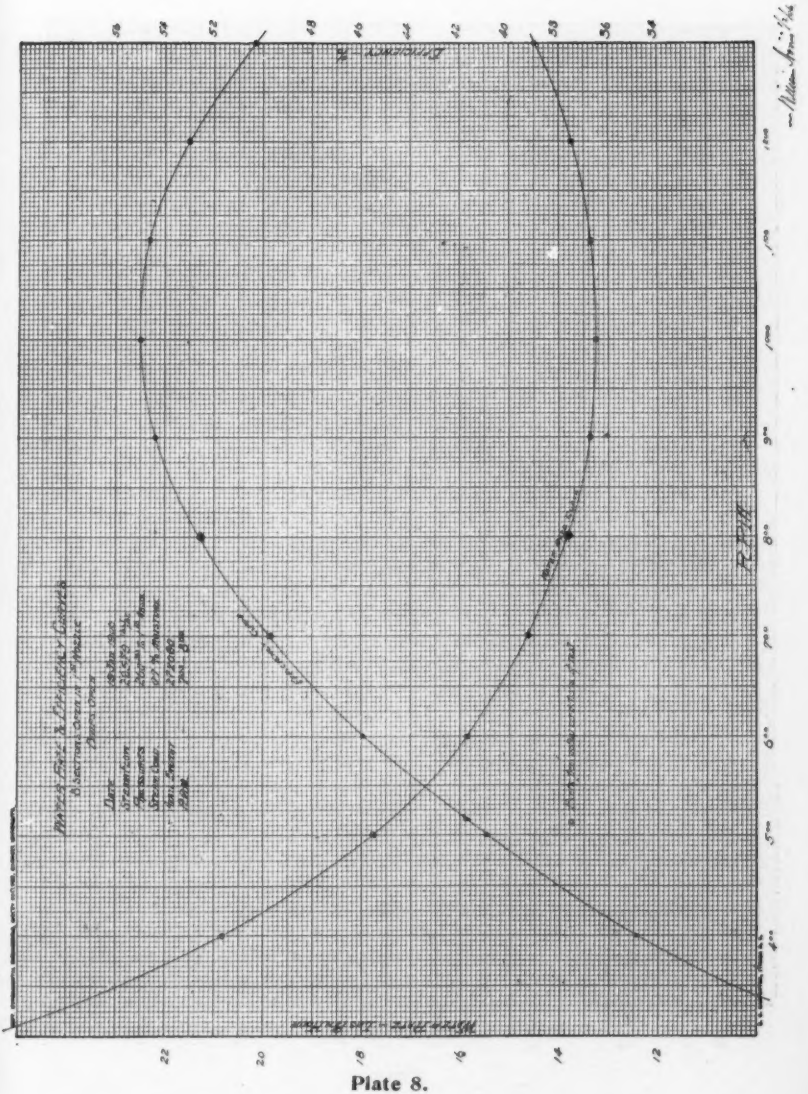


Plate 8.

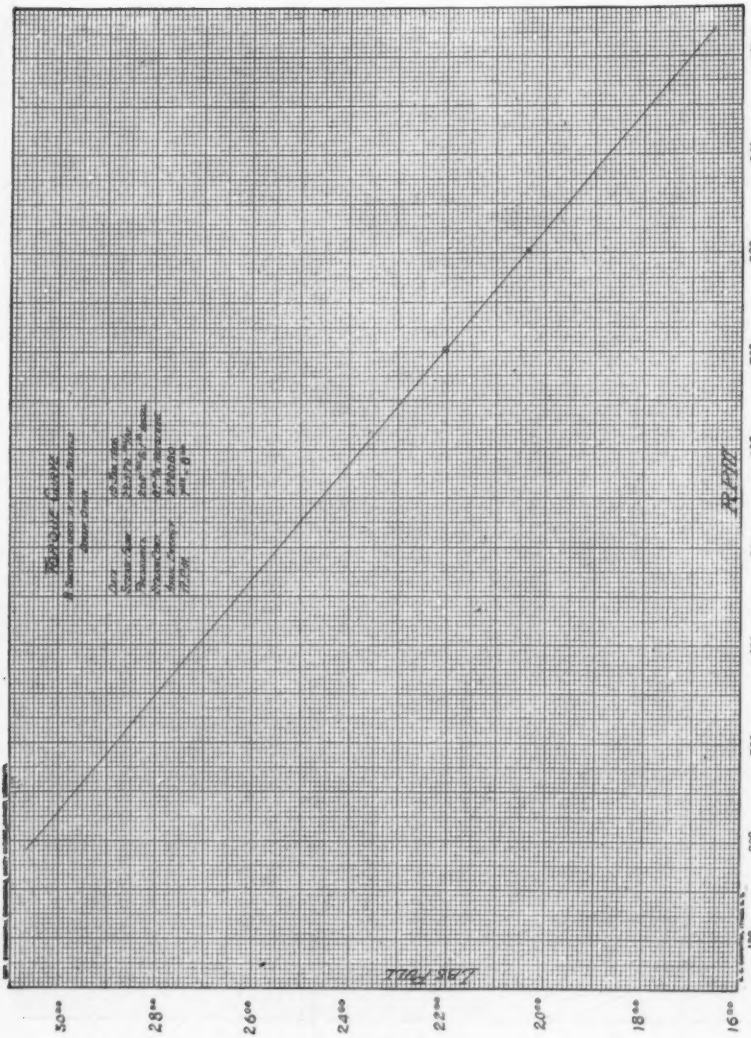


Plate 9.

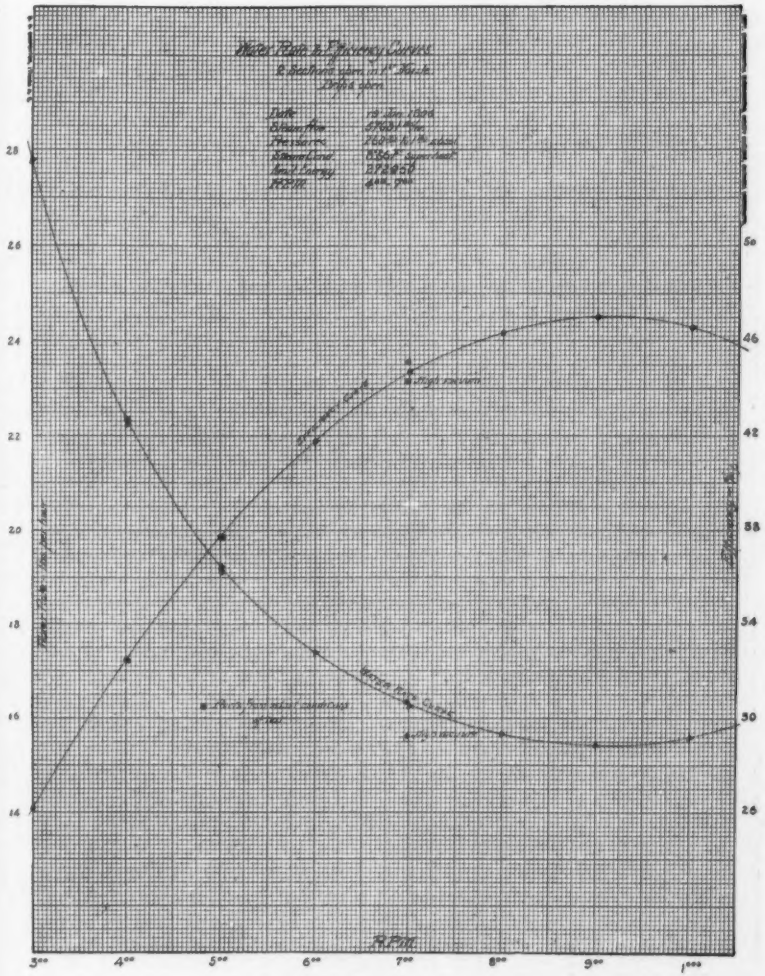

Water Rate & Efficiency Curves
Date
Steam Pressure
Vacuum
Steam Used
Ave. Load
Revs
Water Rate Curve
Efficiency Curve
Revs. per Min.
Water Rate — Lbs. per I.H.P. per Hour
Efficiency — %
I.H.P.
3000
4000
5000
6000
7000
8000
9000
10000
14
16
18
20
22
24
26
28
26
30
34
38
42
46
50

Plate 10.

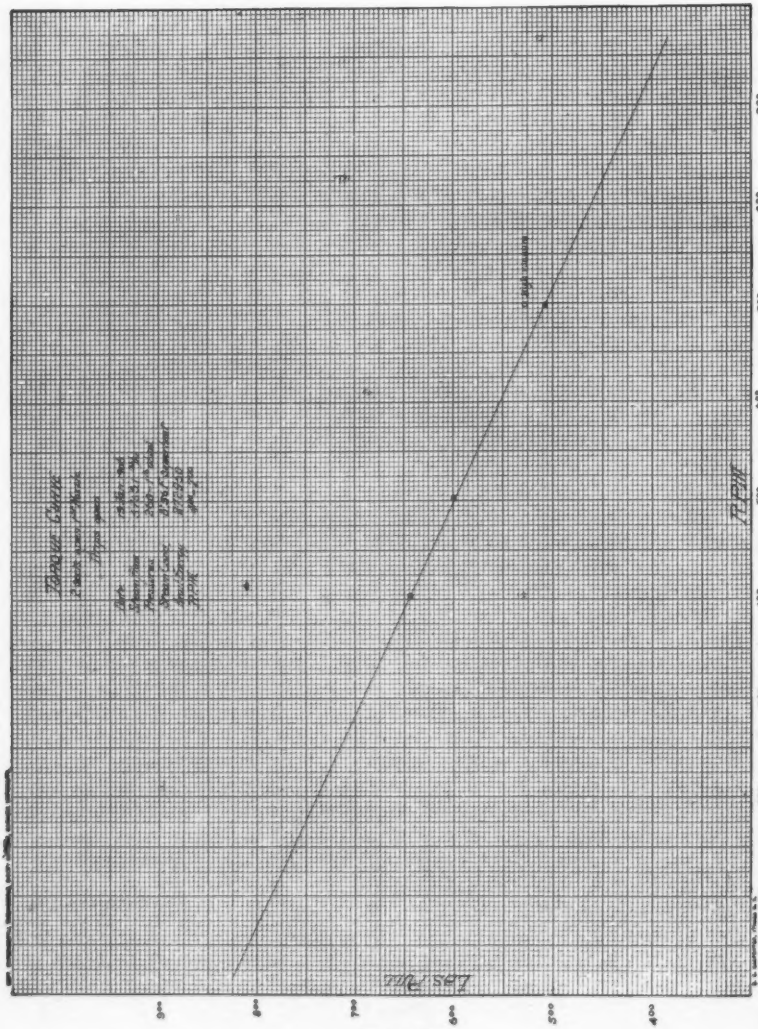


Plate 11.

NOTES.

DEPTH OF WATER AND SPEED OF SHIPS.

The effect of depth of water on the speed of vessels may appear to the average ship owner, and perhaps, to some degree, to the designers of merchant vessels, a subject of remote practical importance; a matter doubtless of interest to the naval architect, but far removed from commercial considerations. No doubt, superficially, this is true; for mercantile vessels run very little in soundings shallow enough to greatly affect their speed. To the designer of war vessels, however, the problem assumes an aspect of considerable importance, for experience has shown how largely the trial-trip performance—on which the acceptance of the vessel depends—is influenced by depth of water. Seamen have long known, as a general fact, that speed is reduced when a ship passes from deep to shallow water: and some time ago the Admiralty laid down a long-distance course off the South Coast upon which to run trials of large ships, it having been found impossible to get desired speeds on the comparatively shallow official mile in Stokes Bay.

It is, however, not only the size, but, to a large extent, the speed of a vessel that influences the problem; a matter the modern theory of resistance would indicate, but which appears not always to have been borne in mind; although, on the other hand, some later writers appear to have lost sight of the influence of size in considering the importance of speed. We owe the fuller recognition of the influence of speed and a closer study of the whole subject, to those wonderful craft the torpedo boats; and if these little vessels never sink another ship, they have justified their existence by the impetus their construction has given to the science of naval architecture in all its branches.

Without referring to the investigations of Scott-Russell and other experiments that have become classic, it may be said that it was the work of Captain A. Rasmussen, of the engineering department of the Royal Danish Navy, which first drew attention to the true influence depth of water, within certain limits, has upon the speed of fast vessels. In an article contributed to our issue of September 7, 1894, and afterwards reprinted in the Transactions of the Institution of Naval Architects,* he gave particulars of trials made with a torpedo boat. The article pointed out that it was usually believed that decrease in the depth of water was always accompanied by decrease in speed; but by the aid of curves of speed and power plotted from the results of trials made in varying depths of water he showed that this was not invariably true, as under certain specified conditions an increase of speed was obtained in shallower water. The article still remains one of considerable interest, and may be said to have been the starting point of a new branch of study in the science of naval architecture. Captain Rasmussen also pointed out in this article the important fact that "the point of inflection on the curves (the 'hump,' as we more inelegantly call it) becomes more marked and occurs earlier the shallower the water." In a paper contributed to the Institution of Naval Architects between four and five years later, Captain Rasmussen gave further valuable and most suggestive results obtained on other trials.† Although the credit of thus inaugurating a valuable and interesting study must be awarded to a Danish engineer, we hope we may claim some small position in the matter for this country, as Captan Rasmussen received his professional education at the Royal Naval College at Greenwich.

The year following the reading of Captain Rasmussen's paper, an Italian member of the same Institution, Colonel Giuseppe Rota, presented another contribution on the same subject, in which he gave the results of certain experiments made in the tank of the Royal Italian Dockyard at Spezzia.

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*See Transactions, Vol. XLI, page 18.

†Ibid, page 12

This is also an extremely valuable contribution to the literature of the subject, and it is to be regretted that Colonel Rota did not extend his remarks by discussing more fully the results he obtained. If he had done so, we think his labors would have been more immediately fruitful, as some students of the subject may have failed to realize how much there was in the paper.

Fortunately, the matter has been taken up by another distinguished member of the Institution of Naval Architects, who has recently given a brief but extremely valuable review of the whole question of the influence of depth of water on speed.

In the Watt Anniversary lecture delivered this year, by Mr. Sydney W. Barnaby, on January 19, at Greenock, the author took for his subject "Marine Propulsion;" and as he brings together the chief results of other experimenters in this field of research, and, moreover, makes practical application of an extremely valuable nature of the data already recorded, supplemented by his own observations, it will be useful to draw attention to his work. Speaking of Captain Rasmussen's experiments, to which we have already referred, Mr. Barnaby points out that depth of water may bear such a relation to the speed and size of a vessel that what is known as a "canal wave" will be generated, and that in a channel of rectangular section the speed of this may be expressed by the well-known formula, $v = \sqrt{g\,h}$. Rasmussen found that though at half-power the speed of his boat was 5¼ knots less in 15 feet of water than in 48 feet, yet when the boat was run at full power the contrary result was observed of the speed, being 1¼ knot more in 15 feet of water than it was in 48 feet of water.

It was the publication of these results which, as we have said, set naval architects thinking, and those most interested to making experiments of their own. Sir John Thornycroft ran some trials in the Thames with a small steel steamer drawing only 10 inches of water, and obtained some remarkable results. In the deeper channel of the river the boat traveled at her normal speed, accompanied by the usual wave phenomena: but on running over a mud flat, where there was only a few

inches of water under the bottom, wave making ceased almost completely, and not only did the speed palably increase, but the boat ran with a smoothness and apparent absence of effort that was quite remarkable.

Mr. Barnaby gave other instances of a similar character. Some first-class Thornycroft torpedo boats were tried upon the Maplin mile, and it was proved by repeated tests that at least 1½ knots more could be obtained at low water with a depth of 50 feet, than at high water when the depth was 65 feet.

The two papers on this subject, read respectively by Mr. Harold E. Yarrow and Mr. W. W. Marriner last year, before the Institution of Naval Architects, will be fresh in the recollection of our readers. They are supplementary to each other, and form together one of the most valuable contributions to the subject yet published. It will be remembered that trials were made with a torpedo vessel over a measured course of varying depth off the mouth of the Thames. These trials were of an exhaustive nature, and must have entailed a large amount of expense. The thanks of all are due to Messrs. Yarrow, who, with their usual liberality, allowed the full details to be made public, for the benefit of everyone, including their competitors.

Before proceeding to deal more fully with the results of these trials—or rather, perhaps, with the valuable deductions Mr. Barnaby draws from them—we may refer to some results obtained by the trials of a German torpedo boat, 200 feet long and 374 tons displacement, as detailed in a paper read before the Schleswig-Holstein Bezirksverein.* The German engineers were fortunate in having a course where the bottom of the sea deepens gradually and uniformly from a depth of 7 meters to 60 meters, and a series of six parallel measured miles were marked off. Trials were made at each of these depths at varying speeds from 12 knots up to 27 knots. Mr. Barnaby has plotted the data obtained on a diagram; but the results may be briefly stated as follows: 1. At depths of 130 feet, and more,

* "Zeitschrift des Vereines Deutscher Ingenieure," December 10, 1904.

the influence of depth was insignificant. 2. At 27 knots the most unfavorable depth was 80 feet. 3. At little more than half power the speed was 5.64 knots less in 23 feet of water than in 200 feet. 4. At full power the speed was 0.82 knot higher in 23 feet than in 200 feet. 5. At 12 knots and under the influence of depth was inappreciable.

Turning to the trials of the destroyers of the River class, the difficulty in getting the contract speed on the usual trial course will be remembered. It was found by trials made by Messrs. Thornycroft with some of these boats that there was a difference of 2 knots when the runs were made in 50 feet of water on the Maplin mile, and in 100 feet depth off Dover, the higher speed being obtained in the deeper water. When the boats were taken to the Clyde and run on the Skelmorlie mile, which has a depth of water of 240 feet, a further increase of 1 knot was secured.

It was also with destroyers of the River class that Messrs. Yarrow made the experiments off the mouth of the Thames, the results of which were given in the papers read by Mr. H. Yarrow and Mr. Marriner. Mr. Marriner attached to his paper a curve embodying the results, and gave the equation $d = \frac{V^2}{11.3}$ as that which denoted the most unfavorable depth at which the boats could be run; d being depth in feet, and V the speed in knots. This, as Mr. Barnaby points out, gives the same curve as would be obtained by the formula for the "canal wave," $V \sqrt{g\,h}$.

Mr. Marriner therefore showed, for the River class of destroyers, what depths of water should be avoided, and the information would be of great value to designers of similar craft; but Mr. Barnaby has been able to put the details in another light, and, by the aid of Colonel Rota's experiments, has deduced what would be the best depths for a number of different types of destroyers. The resistance curve of a high-speed vessel shows a more or less rapid increase of resistance, with increase of speed up to a certain critical speed; but above this critical speed the curve, instead of continuing to ascend more

rapidly, drops again, thus forming the "hump," or point of inflection. The position of the "hump," Mr. Barnaby points out, coincides with the speed at which the length of the wave proper to that speed agrees with the length of the wave that the hull tends to form, a solitary, or non-repeating wave, being formed. The bad influence of this "conjunction of wave-lengths"—noticed by Mr. R. E. Froude†—was plainly shown on Mr. Marriner's curve of worst depths. When the speed of the boat is increased beyond the critical point of conjunction of wave-lengths, the wave can no longer keep pace with the boat, and there is practically no wave making.

Taking the curves obtained by Colonel Rota with his model, Mr. Barnaby plotted the *locus* of minimum resistance, and on the diagram thus obtained he interpolated curves showing the depth at which there would be least resistance for a few vessels of the destroyer type that resembled the model sufficiently in general form to render the comparison fair. From this diagram it appeared that for the coastal destroyers of from 200 to 215 tons, the best depth of water for a speed of 26 knots was shown to be 20 feet; whilst the normal deep-water resistance would be met with at a depth of 98 feet, no appreciable advantage being gained by running at a greater depth. For destroyers of the River class, of from 560 to 600 tons, and 25½ knots speed, the resistance would be at a minimum for shallow water at a depth of about 29 feet, and would then increase until it reached the maximum at 67 feet, after which it would decline until, at 135 feet, it would be the same as at 29 feet. Beyond 135 feet, the resistance would remain constant at all greater depths.

It is extremely satisfactory to learn from Mr. Barnaby that these figures accord very well with results observed on trials of the actual boats. The River class will reach their speed in 25 feet to 35 feet of water, and also at depths above 120 feet. but not at any intermediate depth. Probably they would do better at the 135 feet mentioned, but they seemed to be at their worst in 60 feet to 70 feet of water.

† "Transactions of the Institution of Naval Architects," Vol XLII page 245.

Calculating from the data thus obtained, Mr. Barnaby constructs a curve from which it would appear that a 780-ton vessel of 33 knots—figures that correspond to the new ocean-going destroyers—would do best in shallow water at a depth of about 32 feet, and that in deep water the resistance would be normal at about 150 feet or upwards. A vessel of similar form, but of somewhat larger displacement and of a speed of 36 knots, would be likely to reach its best shallow-water performance at a depth of about 36 feet, whilst about 170 feet of water would be needed to give normal resistance in deep water.

These results are extremely interesting, not less so because they show the value of scientific analysis applied to observed results. Mr. Barnaby has also examined the curves of Mr. Yarrow's paper, and finds that the best depth for the boat tried coincides with that which would be calculated from Colonel Rota's curves—namely, 29 feet. The *locus* of minimum resistance appeared to be constant for the Yarrow model at speeds of from 22 to 27 knots, and, on the assumption that it would remain constant up to a speed of 32½ knots, the best depth for coastal destroyers would work out almost exactly the same as obtained from the Rota model, or 20¼ feet for the boat, as against 20 feet, as calculated from the model; whilst for the 33-knot boat the depth would work out at 31¾ feet from the Yarrow curve, as against 32 feet, calculated from the model experiments of Rota; and for the 36-knot boat the depth would be 35½ feet from the Yarrow curve, as against 36 feet from Rota.

As Mr. Barnaby remarks, the agreement between these figures is remarkable, seeing that the two sets of experiments were quite independent. It will be interesting to see if these deductions are corroborated by the trials of these classes which are yet to take place.

One more point brought out by Mr. Barnaby may be alluded to in bringing our notice of this interesting study to a conclusion. The curves Mr. Barnaby plotted showed that, while the resistance remained practically constant in depths above that which is called the *locus* of minimum resistance in deep

water, the resistance varies very rapidly with small variations of depth on either side of the critical depth in shallow water. The moral of this is that trials should be run in deep water, for it is evidently very difficult, if not impossible, to find a measured mile with a constant depth of water—remembering the variations due to tide—where the soundings are just those required for each different type of ship at each speed. Whether the Admiralty will be able to find a measured mile in the South equal to the Skelmorlie course, with 40 fathoms, is a matter of considerable doubt. It will be noticed, however, that 30 fathoms would be sufficient depth for all the vessels to which reference has been made. In the meantime it may be pointed out that data obtained from steam trials of vessels, made in depths of water other than that required for giving normal results in deep water, are very untrustworthy as a guide for calculations designed to determine the power required for similar vessels of different dimensions or speeds.—"Engineering", February 2, 1906.

CONCLUSIONS AS TO SUBMARINES.

Sir William H. White, former Naval Constructor in Chief of the British Navy, whose valuable articles on submarines published in the "London Times" have been extensively quoted in these columns, has concluded the series with a careful summary of his general conclusions. The following extracts from his final article are of general interest:

The operations at Port Arthur and the battle in the Korean Straits have demonstrated conclusively the power of swift torpedo vessels, even in daylight, when handled by gallant and skilful seamen and have given fresh illustrations of the tactical advantages attaching to high speed. It cannot be overlooked that while our latest submarines have a surface speed of thirteen knots, torpedo vessels of not much greater displacement and about the same cost have more than twice that speed, associated with greater radius of action, superior sea-keeping qualities, an equal torpedo equipment, and an armament of quick-firing guns.

Under circumstances when submarines would be of little or no fighting value—as, for example, against ships proceeding at good speed in a considerable sea—surface torpedo vessels would remain efficient. The true conclusion, therefore, is that submarines are not and cannot be substituted for torpedo boats and destroyers, but are complementary thereto. It may be doubted whether, on the whole, swift surface torpedo boats may not prove more effective than slow submarines in getting torpedo attacks "home" and injuring an enemy even in daylight. Great risks must be taken, no doubt, but the chances of escape seem to be greater than would be anticipated from *a priori* considerations, especially when attacks of torpedo craft are supported by other vessels of the fleets to which they are attached.

During peace maneuvers, when ships on both sides belong to the same fleet, the danger of making mistakes is greatly increased; and possibly experience gained on such occasions has tended to exaggerate the estimate of probable mistake in actual war. With an efficient intelligence department an Admiral would be well informed as to the whereabouts of the enemy before he launched his torpedo flotilla on a night attack, and would not be likely to have his own large ships in the immediate locality.

The broad conclusions of French writers, based on their own experiences as well as on British experiments so far as they are known, may be summed up in a sentence—the best method of meeting submarine attacks consists in close watch, the use of attendant small craft, and mobility of target. Blockading squadrons must keep their main strength at a considerable distance from bases of operation of submarines, and must be protected by inshore squadrons of small, swift vessels. French writers are disposed to make light of experimental British methods for dealing with submarines by the employment of specially equipped destroyers or vedette boats.

One possible means of ascertaining the approach and bearing of submarines by ships attacked, or of enabling submarines to locate an enemy, is found in a system of signals developed in

America, and described in some recent articles. As this system enables sounds to be heard and vessels to be located at a distance of six to eight miles, the approach and bearing of a submarine should be ascertainable in most cases while she is still at a considerable distance; since she would approach within a comparatively close range of her enemy before diving, and up to that time would usually be propelled by gasoline engines, which produce sounds of an unmistakable character. Trials made in the United States confirm this opinion, but only exhaustive experiments can decide the question.

The universal law affecting naval construction has proved true with submarines. Within four years British vessels of the type have increased in displacement from 120 tons to 300 tons. The power for surface propulsion has increased from 150 to 850 horsepower; surface speed has been raised from seven and one-half knots to thirteen knots, and the radius of action has been practically doubled. There is now a disposition abroad to "hark-back," and to build vessels of smaller dimensions suitable for coast and harbor defense or for lifting and carrying on shipboard. If this is done, there is practical certainty that the law of increase in size will affect these specialized coast and harbor-defense vessels and history will repeat itself. If the idea of giving gun armaments to submarines is realized, then, for reasons previously explained, still further increase in size and cost will be inevitable. There is no finality in the design of war vessels, and it is idle to assume that the last word has been said as to possible types of ships or of propelling apparatus.

A proposal which has been seriously considered and subjected to experiments is the substitution for submarines of controllable torpedoes (such as the Brennan) operated from swift small vessels. In this manner the attack could be delivered from a considerable distance; the vessel from which the torpedo is operated would take up her position quietly, so as to avoid discovery, and at a distance where she would be difficult of detection, especially at night. Against ships at anchor such an attack should be successful, so far as the attainment of station-

ary targets by torpedoes is concerned, provided the vessel serving as a base was not injured seriously. As the result of trials it is claimed that similar success can be obtained against vessels in motion and rapidly changing their course, but only limited trials have been made.

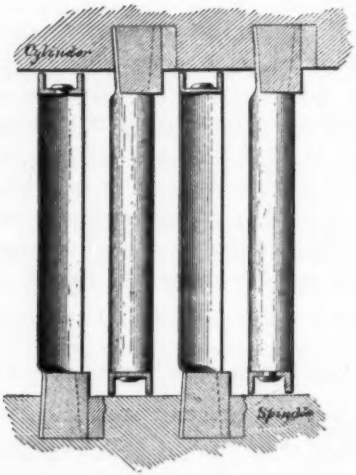
The latest Admiralty return (Sir Charles Dilke's) shows France to possess thirty-seven completed submarines and thirty-two to be building; as against seventeen completed and twenty-three building for Great Britain. Of those building, twenty-two French and eleven British are to be laid down this year.

Having regard to our greatly superior productive power and the possibility on an emergency of rapidly multiplying approved types, as well as the great advances which are being made in mechanical engineering, it is obviously undesirable for this country, in peace time, to undertake the simultaneous construction of a considerable number of submarines. Better results can be achieved by gradual additions, as successive designs may embody improvements based on experience and invention.—"Army and Navy Journal," Dec. 9, 1905.

AN AMERICAN MODIFICATION OF THE PARSONS STEAM TURBINE.

The steam turbine recently installed by the Allis-Chalmers Company at Utica, N. Y., for the Utica Gas and Electric Company has aroused a great deal of interest. The turbine is rated at 1,500 kilowatts normal load, and is direct-coupled to a two-phase, sixty-cycle, revolving field alternator, which it drives at a speed of 1,800 revolutions per minute. The interest in the new turbine is due not to any new principle of operation, for in this respect it closely follows the Parsons type, but rather to certain constructional features which mark a distinct advance in turbine building. The improvements relate particularly to the manner of assembling the blades and securing them to the cylinder and spindle, also to the novel method of reinforcing and protecting the tips of the blades.

The illustration is a section taken through a portion of the cylinder and spindle, showing the blading construction. The blades are formed with dovetailed roots, which are fitted into slots cut in base rings. These rings are also of dovetail shape in cross section, and are inserted in dovetailed slots cut in the cylinder and spindle respectively, and are secured by key rings in the manner of a lewis bolt. To hold the key rings in place, the slots are undercut, and after the key rings have been driven into position, they are upset into the undercut grooves. The tips of the blades are reinforced by shroud



rings of channel form. The blades are secured to these rings by means of shouldered projections, which are inserted in slots in the rings and riveted over. The slots are uniformly spaced and formed at an angle to position the blades at the proper working pitch. All of these operations are performed by machinery, insuring an absolute uniformity of blading. The shroud rings, aside from bracing the ends of the blades and preventing any individual blade from working loose, serve also to prevent stripping of the blades in case of contact between rotating and stationary parts. This has been one of the chief

difficulties encountered in previous constructions, a difficulty which inventors have long been striving to overcome. The use of shroud rings also permits a much smaller working clearance, reducing loss by leakage of steam past the ends of the blades, a loss which has been very serious in previous constructions.

The blading is assembled in half rings, and carefully tested before being set in position in the turbine. The outwardly-projecting flanges are then turned and bored to provide the necessary working clearance. Owing to their channel form, the shroud rings afford ample stiffness to the construction. Yet the flanges are quite thin, so that if for any reason one of the moving shroud rings on the spindle should come into contact with the stationary cylinder, or if one of the stationary rings on the cylinder should accidentally touch the moving spindle, the friction would not develop a dangerous degree of heat.

The criticism of this method of blading has been made that the blades are weakened at the root, owing to the dovetail formation. In answer to this, it is claimed that while the cross-sectional area at this point is slightly less than the normal section of the blade, yet owing to its shape the blade is really stronger at the root to resist cross breakage than elsewhere. An accident recently occurred at the West Allis shops which served to show the strength of the turbine blading. A turbine spindle with blades assembled was put into a lathe for the purpose of turning off the shroud rings to give the necessary working clearance, as well as to smooth up the key rings which hold the blading in the dovetail grooves. The workman who was running the tool down between two rows of blades accidentally moved the tool rest too far, so that the back of the tool ran into one of the rows of blades. As a result the blades simply bent over as far as the lathe tool could push them, and the channel shroud ring was distorted by reason of its fastening to the blades, but not a single blade pulled out or broke off at the root, and not a single blade pulled out of the shroud ring or even broke off at that point, where the cross section is less than one-third of the cross section of the root. It was

necessary to cut away the shroud ring with a hack saw before the blades could be removed for the purpose of inserting new ones.

Another feature of this turbine which is likely to arrest attention is the fact that only two balancing pistons are shown. In previous constructions of this type three pistons are used. As a matter of fact, there are three pistons in the present construction, the third one being applied at the low-pressure end, and being concealed in our photograph behind the large end of the spindle. The advantage of applying the piston at this point is that it relieves the shaft of undue tension, for it will be evident that the greatest axial pressure on the spindle is exerted at the low-pressure end. The piston can also be made smaller than in previous constructions. Instead of using "dummy packing" on these pistons, a packing of radial baffling type has been adopted. That is, the peripheries of the pistons are grooved and fit into grooved bearings in the cylinder casing. In this manner small axial clearance in the turbine is eliminated.—"Scientific American," February 10, 1906.

THE DEVELOPMENT OF THE CAPITAINE PRODUCER-GAS MARINE ENGINE.

The application of the Capitaine producer-gas marine engine the possibilities of which have been demonstrated in a very practical manner by the Thornycroft Shipbuilding Company, is being extensively developed in Great Britain. So far the utilization of this system of generating power has been confined to small vessels, the Thornycroft Company having only completed arrangements with the inventor to this end. The rights to manufacture and apply the invention to large vessels have been acquired by Messrs. Beardmore & Co., Ltd., who are carrying out some interesting experiments to this end.

This firm has under contruction an installation of 500 horsepower, and this is by far the largest engine of this type that has been so far undertaken. Under these circumstances the

construction of such an engine presents several problems that have not had to be encountered in the case of the smaller powered plants. Yet there is a great opening for this type of engine, not only for propelling purposes, but for auxiliary machinery purposes, especially where weight of machinery and economy in coal consumption are of vital importance. The total weight of such engine is about 75 per cent. of the ordinary engines and boilers, while the fuel consumption is 50 per cent. less.

This 500-horsepower plant will comprise five cylinders of the vertical type, working on the Otto cycle principle, and fitted with a suction gas producer specially designed to work continuously with ordinary bituminous coal. The framing of the engine is of plate steel exclusively with the exception of the cylinders, which are the only large parts of cast iron. By means of this design a rigid yet light construction is obtainable.

The builders have also designed a new type of reversing gear, which possesses several interesting and novel features. This gearing will enable the engine to run in either direction and dispense with feathering propellers, bevel gearing, and so forth for reversing. The gas producer is also being arranged so as to supply the engines with gas free from tar, and is to be fired by a mechanical stoker so designed as to prevent the formation of clinker when using bituminous coal.

When this engine is completed, it will first be submitted to exhaustive and severe trials on land under conditions resembling as near as possible those obtained at sea. If they should prove satisfactory, it will then be installed upon a Glasgow coasting steamship, and the firm will then proceed with the construction of an engine of 1,000 horsepower, for which the designs have been prepared, but the construction of which has been postponed until the results of the 500-horse power engine have been obtained, so that any disadvantages or weak points that may develop may be eliminated from the larger engine. The engine now in hand will be completed and tested within the next two months.

That there are great possibilities for this system of generating power is evidenced by the practical interest that is being shown in the invention by the British Admiralty, who are also constructing a high powered engine at Manchester, and which will also be subjected to a series of severe trials by the naval engineers. If the invention can be proved to be as successful in the larger sizes as it has been in smaller craft, it has a great future as the power generator in large vessels, and especially in bucket dredges and towing craft, where a maximum of power is desired at the minimum of expense, both as regards initial cost, space occupied, weight, and economical coal consumption.—"Scientific American," Jan. 6, 1906.

THORNYCROFT 120-HORSEPOWER GASOLINE-PROPELLED TORPEDO LAUNCH.

An interesting development of the application of gasoline motors to small naval vessels has been carried out by the Sir John Thornycroft Company, Ltd., the well-known torpedo boat constructors at Chiswick-on-Thames. This is a gasoline launch carrying a 14-inch Whitehead torpedo. The launch has a length of 40 feet over all, with a beam of 6 feet 2 inches, draught of 2 feet 7 inches, and a displacement of 4½ tons. The hull is constructed of galvanized mild steel, and the craft lies very low in the water, thus affording but a small target to the enemy. A turtle deck is fitted forward and is continued over the engine in a portable piece, extending as far as the after end of the engine, where on the port side is placed the steering wheel and reversing lever. A watertight bulkhead is provided, so that in case of damage to the stem the boat will not sink. The stern of the vessel is made very broad in order to overcome the difficulty of stability when launching the torpedo over the side. Further, in order to prevent the splash from the bow wave being blown inward, "whiskers," or detachable spray boards, are fitted.

The propelling engine consists of a four-cylinder Thorny-

croft gasoline motor of their standard marine type. The stroke is 8 inches, with a bore of 8 inches. The engine is of the lightest construction consistent with the maximum of strength, and is so well balanced that at 900 revolutions, at which the 120 brake horsepower is developed, there is only the slightest trace of vibration. The cylinders are of cast iron, with the water jackets cast round them. They are bolted directly to the aluminum bedplate. The pistons are of cast iron of light construction, with cast-iron packing rings. The connecting rods are of special stamped steel, with steel gudgeon pins and white-metal bearing surfaces. Owing to the fact that all the reciprocating parts are made of the very highest class of material, it is possible to reduce the weight to a minimum, with an attendant appreciable minimizing of vibration. Both the inlet and exhaust valves are mechanically operated by cams and tappets, while in order to reduce the number of spare parts, the valves are made interchangeable. The cam shafts are arranged in the crank chamber, so that they are well lubricated by the same splash arrangements as are provided for the other moving parts.

There is a centrifugal apparatus on the cam shaft for governing purposes. This is connected to the throttle valve by means of a dash-pot arrangement, which prevents "hunting" when the engine is running light. Cooling of the cylinders is effected by means of a pump. Electric high-tension ignition is fitted. Reversing and stopping are effected by means of a Hele-Shaw friction clutch. The utilization of this type of clutch enables instantaneous maneuvering to be carried out, without any shock, while it is not subject to the great want of efficiency which is inherent in all reversing propellers. The engine itself only weighs 25 hundredweight complete, which is equivalent to only 23.25 pounds per brake horsepower. The engine is arranged to run on either gasoline or the heavier fuel, paraffin. Owing to the care exercised in the design of the vaporizer, and the explosive mixture arrangements, the engine is rendered very economical in running, the consumption in the case of gasoline averaging less than one pound per

brake horsepower per hour. The capacity of the fuel tank is 100 gallons, this being sufficient for a run of ten hours. The motor is started by means of compressed air contained in a reservoir placed in the stern of the launch. The compressed air is obtained from a Brotherhood compressor driven by a 6-horsepower single-cylinder Thornycroft gasoline motor. The exhaust gases are carried into the outer atmosphere through the funnel.

The 14-inch Whitehead torpedo is carried in the after part of the boat, and is launched by lowering over the side by means of side drop gear, the body being first directed bow-on to the object which it is desired to hit.

The launch has a speed of 18 knots per hour, and should prove a convenient and useful acquisition to a battleship. In view of its small dimensions, it could be easily stowed on the deck of a battleship, and quickly launched when desired. At the same time, owing to the small target it offers to hostile guns, and its silence in running, it should prove a formidable antagonist, being able to approach an enemy, launch its torpedo, and escape. The British Naval Department, realizing the possibilities of gasoline-propelled launches of this description, propose carrying out a series of tests with them to ascertain their possibilities and efficiency under war conditions. —"Scientific American," Dec. 30, 1905.

THE NEW TURBINE TORPEDO OF THE UNITED STATES NAVY.

It is always difficult to ascertain just what other navies are doing in torpedo work, because special secrecy is maintained with regard to what is still considered to be one of the deadliest forms of naval warfare; but the new turbine torpedo, known as the Bliss-Leavitt model, which has recently been adopted by the United States Government, furnishes the American Navy with what is probably the speediest and most effective weapon of the torpedo type in existence.

The new weapon conforms, in its external appearance and in the leading features of its internal sub-division and method

of control, to the Whitehead, but in size, power, speed, range and accuracy it far surpasses it. The Whitehead of the standard type as used in the United States Navy has a speed of 28 knots and a range of about 1,200 yards, and about 22 knots at 2,000 yards. The new torpedo has a range, guaranteed by contract, of 3,500 yards, and its speed is 28 knots at this range and 36 knots at 1,200 yards range. The United States Government has been so well satisfied with the new weapon that contracts, amounting to several millions of dollars, have been awarded for the construction of this type of torpedo, which, from this time on, will be the only type used in the Navy. Two sizes are being made: one, 18 inches in diameter, which can be fired from the existing 18-inch tubes on our battleships and torpedo boats; and the other, a much larger and more powerful torpedo, 21 inches in diameter. The 18-inch torpedo of the new type has an effective range of 2,000 yards and a speed of 33 knots, and 100 of this type have been contracted for, while of the larger 21-inch, 300 are called for by the contract, Thirty of the 18-inch and two of the 21-inch have been delivered at the torpedo station at Newport, where officers and men are instructed in torpedo work under probable war conditions.

The new 21-inch type consists essentially of three sections. First, the head containing the explosive; then the central flask in which the compressed air for driving the torpedo is stored; and last, the after body, which contains the turbine for operating the propellers, the immersion chamber for regulating the depth of the torpedo beneath the surface of the water, and the gyroscope gear by which the torpedo is automatically steered and maintained on its proper line of flight.

The head is a beautiful specimen of hammered sheet-metal work. It is formed in two halves, divided longitudinally, the edges of the joints being made with a square, saw-tooth form and brazed together. The war head, which, as distinguished from the practice head, is used only in actual hostilities, is loaded with 132 pounds of guncotton, containing 25 per cent. of moisture. The guncotton is packed in disks through the

center of which is a hole that contains a cartridge primer of dry guncotton for detonating the charge. The small propeller carried at the extreme point of the torpedo is for preventing premature explosion, which it does by locking the firing pin. When the torpedo enters the water, the revolution of the propellers releases a sleeve, which uncovers the firing pin, putting it in position to strike the detonating primer the instant that the torpedo finds its mark. The central body, or shell of the torpedo occupies a little more than one-half the total length. It is made of a special forged steel of an elastic limit of at least 90,000 pounds. The rough forging is over 1¼ inches in thickness, and it is bored and turned down in the lathe to a finish thickness of seven-sixteenths of an inch. The "flask," as the central portion, or air chamber, is called, is to the turbine engine of the torpedo what the boiler is to the reciprocating engine of a steamship. It is charged at an initial pressure of 2,225 pounds to the square inch.

The after portion of the torpedo, or the tail, contains in its forward end the wonderful little torpedo engine which drives the propeller. It is of the Curtis compound type, and consists of a central row of fixed blades and two wheels, one 11⅛ inches and the other nearly 12 inches in diameter. There are two propellers, adapted to run in opposite directions, one being fixed upon the central shaft, and the other upon an enveloping outer shaft. The turbine runs at a speed of about 10,000 revolutions per minute, which is reduced by suitable gears to a speed of 900 revolutions for the propellers. At this speed the turbine developed about 160 horsepower, the corresponding speed being 40 knots an hour, although the contract speed required by the government is only 36 knots.

Immediately astern of the compartment containing the turbine is the wonderfully ingenious and delicate apparatus for maintaining the proper depth of immersion and for steering. The regulation of the depth is effected by means of a vertical diaphragm, on one side of which is the water, which is allowed to enter by holes provided in the shell for that purpose, and on the other side a series of coiled springs, the water pressing

against the diaphragm on one side, and the springs pressing the diaphragm in the opposite direction on the other side. The springs are adjusted so that their pressure shall exactly equal the pressure of the water at the given depth at which the torpedo is to travel. If the torpedo descends below that depth, the water pressure, overcoming the spring pressure, pushes the diaphragm inwardly. If the torpedo is above the desired depth, the springs overcome the water pressure, and push the diaphragm outwardly. The center of the diaphragm is attached to certain levers and rods, which pass through the tail of the torpedo and act on a pair of horizontal rudders, throwing them up or down, according as the diaphragm is pressed inward or outward, and thus correcting the deviation of the torpedo from the horizontal plane at which it is designed to travel.

Astern of the immersion chamber is located the steering gear. This is a modification of the principle employed in the Obry gear, and depends upon the well-known tendency of a gyroscope to maintain itself in its original plane of rotation. The Obry gear was given its high velocity by means of a coiled spring which was released at the moment of firing. In the Bliss-Leavitt torpedo the spring is dispensed with, and a small reaction turbine is used in its place. This consists of a disk with a series of discharge orifices arranged tangentially to the circumference, which are fed with compressed air. The air rushing from the orifices reacts on the disk, and turns it exactly in the same way as did the pipes on Hero's original turbine of two thousand years ago. If the torpedo tends to deflect to the right or to the left, this little gyroscope turbine maintains its original position, and its angular motion with regard to the torpedo (or to speak more accurately, the angular motion of the torpedo about the gyroscope) serves to actuate a very ingenious mechanism, which turns the vertical rudders to the right or left, and corrects the deviation. The turbo-gyroscope is driven at a speed of 18,000 revolutions per minute.

Of course, the most interesting feature in the building of the new torpedoes is the construction of the wonderfully efficient little turbine engine that drives them. The Bliss Company has

designed a very effective machine for cutting the buckets of the turbine wheels. The whole wheel is made out of a single disk of steel, the buckets being integral with the wheel. The machine for cutting the buckets resembles a double-spindle lathe. The work is held in a horizontal position on the tail stock, and two cutters alternately advance toward the rim of the wheel, make a cut of the desired curvature and recede, leaving the wheel free to revolve sufficiently to bring the next bucket into position for another cut. One cutter operates on one wall of the bucket, and the other on the opposite wall. The result is a wheel of perfect form, carrying a highly finished surface. It should be mentioned here that the remarkably high efficiency in speed and range of the new torpedo is due to the use of a superheating process applied to the compressed air. This consists of a flame which is automatically ignited, the instant the torpedo is launched from the tube, and which burns during the entire run. The compressed-air flask contains a burner or pot, the flame of which is fed automatically with alcohol. The flow is so regulated that an even and steady temperature is maintained in the air flask.

During the past few months, the company has been carrying out a series of very exhaustive tests on board the proving steamer *Sarah Thorpe,* which is anchored in the secluded waters of Noyak Bay, near Sag Harbor, Long Island. Here each torpedo is tested and brought up to the required standard of efficiency in speed and range before being turned over to the torpedo station at Newport. The Navy Department assigned a lieutenant and several gunners to witness and record the run of each torpedo. The target is a submerged net, 100 feet in length, which is located 1,200 yards from the point of fire. The torpedo breaks through the meshes, and after each shot the net is hauled up, and the exact striking point is located by the tear in the net. The maximum deviation in the range allowed is 15 feet to the right and left of the bull's eye, and 30 inches above and below at five feet of depth. Each torpedo must come within these measurements in three out of five trial runs, in order to be accepted. The average speed of the run

is 36 knots, and the time run is about 60¾ seconds for 1,200 yards. The cost of the 18-inch torpedo is about $5,000, and the 21-inch torpedo costs proportionately more.—"Scientific American," January 6, 1906.

THE TURBINE AND THE PRODUCER-GAS ENGINE.

A review of progress in the merchant marine naturally leads us to a consideration of the steam turbine, and its coming rival the producer-gas engine; for although the steam turbine received its first practical application in an electric-lighting plant, its prominent introduction to the world was made in that phenominal little steam yacht the *Turbina.* By the close of the year three vessels will be in the transatlantic service whose successful operations will have set at rest all question as to the availability of the turbine as a drive for ships of the largest class. The *Victorian* and the *Virginian* of the Allen line, each of which is 530 feet in length, are driven by turbines of 10,000 horsepower. The first named made 19, and the second vessel 19½ knots on the trial trip, and they have broken all existing records over their own transatlantic route. The *Carmania,* whose turbines are of over 21,000 horsepower, has made 20.4 knots on trial. There was trouble during the early voyages of the *Victorian* but both of the Allan line ships are now giving excellent service. The 3,000-ton turbine-propelled ship *Loongana* gave a further demonstration of the reliability of the turbine in the course of a thirty-and-a-half day trip from Glasgow to Australia, when an average speed of 15 knots an hour was maintained throughout. This is the longest journey ever made by a turbine-propelled vessel. In the smaller classes of vessels, such as those used for river and cross-channel service, the turbine has scored a signal triumph over the reciprocating engine. A comparison of the *King Edward* with a vessel similar in every respect except that of motive power, showed that while during eighty days of sailing at an average speed of 18½ knots, the steamer with

reciprocating engines consumed 1,909 tons in steaming 12,106 knots, the *King Edward* with turbine engines consumed only 1,429 tons in steaming 12,116 knots. Another absolutely reliable basis of comparison has been afforded by the four new Isle of Man passenger boats, two of which have turbine, and the other two reciprocating, engines. The turbine steamer *Manxman* in the comparative trials steamed 20.3 knots on the same amount of fuel as was consumed by the reciprocating-engine steamer *Antrim* when making 19.5 knots. Five gallons of lubricating oil was saved on each 60-mile trip, and the staff of oilers was reduced from four to two. There was a saving of weight in the turbine engines of 115 tons, or 6 per cent. of the total weight of the ship. Regarding the development of the steam turbine for stationary service, it is sufficient to say that the excellent results achieved in steamship service have been repeated in the great electric-light and power plants on shore. For reasons which we cannot now go into, the stationary turbine has not shown the same superior economy as the marine turbine over the reciprocating engine; but the advantages in small space occupied, small amount of lubrication, and smaller staff required have been decisive.

An event which in importance is probably destined to rank with the advent of the turbine steamer *Turbina* was the appearance during the year of a successful steamship driven by a producer-gas engine. The system was devised by Herr Emil Capitaine. This little vessel, named after the inventor, is 60 feet in length, and is driven by its producer-gas engine at a speed of 13 miles an hour. During a ten-hours' run at 13 knots speed, only 467 pounds of anthracite were consumed at a cost of $1.08. Following the producer-gas steamer we may look to see a producer-gas locomotive—this last an entirely feasible proposition. The claim of the producer-gas engine makers that a properly constructed plant will run on less than a pound of coal per horsepower per hour has recently been vindicated in a series of tests carried on by the Highland and Agricultural Society, in Great Britain, when ten complete plants were submitted for approval. Trials were made of

plants of 20 and others of 8-horsepower capacity. The coal in pounds per brake horsepower per hour consumed by four 20-horsepower plants was respectively 0.93, 0.77, 0.80, and 0.83, and in four plants of 8-horsepower it was respectively 1.22, 1.13, 0.84, and 1.35, all of these being full-load trials. Regarding the prospects of the production of a successful gas turbine not much can be said; for the matter has not, as yet, progressed far beyond the experimental stage—"Scientific American."

FLUE-GAS ANALYSIS ON A WATER-TUBE BOILER.

The subject of flue-gas analysis has attracted considerable attention of late, and attention may therefore be called to the results of a series of tests which were reported in this journal some years ago. They were made to show the character of the gas in a hand-fired water-tube boiler with plain furnace and ordinary setting under certain conditions of running, the test being made with an Orsat apparatus. The boiler was one having the ordinary vertical flame plates, and three vertical passes for the gases, being of the 250-horsepower size. The width of the space between the two side walls was 8 feet 6 inches. The sampling pipe was introduced between the third and fourth row of tubes, counting from the top in the rear pass, a hole being drilled through the side cleaning door for the purpose. The pipe was of the ⅜-inch size, 4 feet in length, and it was perforated with seventy-two $\frac{1}{16}$-inch holes, arranged in three rows, twenty-four in each row. This was laid on the tubes beginning at a distance 12 inches from the inside face of the outer wall, and connection with the gas apparatus was made through a ¼-inch pipe. In this position the sampling tube drew from only one-half of the entire space, but this seems to be a proper arrangement considering that the boiler had two fire doors, and a sample obtained from one-half of the entire width of the space should give a fair indication of the whole. The damper was set at a fixed point so as to give a draft of about ¼ inch water pressure, and the rate of

capacity developed by the boiler was maintained constant so far as this was controlled by the draft. The coal used was a grade of New River semi-bituminous, and the thickness of the bed of coal on the grate at times just before firing was from 8 to 10 inches. The samples were drawn for periods of one minute, and they were taken twice during the period while one firing was being burned out; the first sample being taken two minutes after firing, and the second some twenty minutes, the fire in the meantime having been loosened with a slice bar and sometimes leveled off. The firings occurred at intervals of about thirty minutes.

The interest in these analyses lies in the effect which is produced by the different conditions of the fire. On the first test, which was made one minute after firing 20 shovels of coal, the carbon dioxide was 12.4 per cent., the oxygen 4.4 per cent., and the carbonic oxide 1.1 per cent. After an expiration of 28 minutes, during which time the fire had been barred up, the second test was made and showed that the carbon dioxide had fallen to 10.7 per cent., the oxygen had increased to 9.3 per cent., and the carbonic oxide had fallen off to one-tenth of one per cent., thus showing an increase in the amount of surplus air, and a marked improvement in the combustion due to the reduced quantity of volatile gases as the fire becomes burned down. The third test was made two minutes after firing 32 shovels of coal, and it showed for carbon dioxide 14 per cent., oxygen 2.1 per cent. and carbonic oxide 2 per cent. The new coal added on this firing was an excessive quantity, and this was revealed at once by an increased amount of carbon dioxide and a reduced percentage of oxygen in the gases. The next test was made 24 minutes later, the sample being drawn one minute after the fire was leveled off. The effect of this operation is to fill any holes or low spots in the bed of coal and make it a more solid mass. This test showed 16.9 per cent. carbon dioxide, 1.2 per cent. of oxygen and 1.3 per cent. carbonic oxide. The increased solidity of this mass of coal is shown in the small quantity of oxygen, and again in the continued presence of carbon

dioxide, although somewhat reduced. On the fifth test a sample was drawn two minutes after firing 20 shovels of coal, and we have almost a duplicate of the previous test, the result being carbon dioxide 13.7 per cent., oxygen 2.2 per cent., and carbonic oxide 2.5 per cent. At the expiration of 21 minutes from this time the sixth drawing was made and this occurred five minutes after barring up the bed of coal without leveling it off. The result of the analysis, then, is very different from the corresponding time on the previous test, the carbonic oxide disappearing altogether, the carbon dioxide being 13 per cent. and the oxygen 6.4 per cent. The effect of the more open condition of the coal bed in allowing a greater supply of air to pass through, and, consequently, to increase the oxygen in the gas, is plainly revealed. The seventh test was made two minutes after firing 20 shovels full of coal, and showed carbon dioxide 14 per cent., oxygen 2.7 per cent. and carbonic oxide 1.7 per cent. At the expiration of 15 minutes the eighth sample was drawn, this being done six minutes after the fire was opened with the slice bar, but not leveled off. The quantities then changed, the carbonic oxide disappearing altogether, the carbonic dioxide falling to 11.6 per cent. and the oxygen rising to 8 per cent.

These analyses furnish still further proof, if such were needed, that it is objectionable to carry heavy fires and to fire large quantities at a time. In one of these tests the carbonic oxide amounted to 2.5 per cent., and this loss of unconsumed carbon represents for the time being about 1,500 B. T. U. for every pound of coal consumed. If this boiler worked with an efficiency of 70 per cent., on coal having a calorific value of 13,500 B. T. U. per pound, the loss from this cause represents about one-sixth of the heat utilized. The effect of a thick bed of coal, on the other hand, is in one sense advantageous, for, as will be seen from these examples, it reduces the quantity of free oxygen in the gases, which is an indication that the corresponding supply of air is likewise reduced. In one case the oxygen drops down to 1.2 per cent. and in another to 1.7 per cent. This reduction is accompanied, however, by

a considerable quantity of unconsumed carbon, so that in cases of hand firing, it results in no benefit.—"The Engineering Record."

HOWDEN'S FORCED DRAFT.

A noteworthy fact in connection with what may now almost be termed the "triumphal march" of the steam turbine, as applied to ship propulsion is that practically all the turbine steamers so far produced have been fitted with boilers in which the combustion is controlled by Howden's forced draft. In these steamers the enhanced economy attained over the few which have been fitted with closed stokehold, or other type of forced draft, has been so pronounced as to make it certain that the application of this system of hot-air forced draft in itself places turbine engines in a most favorable position as respects economy compared with well-designed triple or quadruple reciprocating engines. Both the latest Cunard vessels—the *Caronia* and *Carmania*—on which at present exhaustive experiments are being made as to the comparative steam consumption of the two types of engines, the ordinary reciprocating engines in the first, and the turbine engines in the last named, are fitted with duplicate boilers worked with the Howden forced draft. Both vessels, therefore, have boiler plants of the highest efficiency, whereby means are afforded of making a thoroughly accurate comparison. The two new 25-knot express steamers, now building for the Cunard Line, are also fitted with this system, as well as the new Canadian Pacific express steamer *Empress of Britain,* recently launched by the Fairfield Co., and her sister still on the stocks, the *Empress of Ireland.* It is now also being adopted largely in express channel steamers—the Ostend-Dover turbine steamer *Princess Elizabeth,* for example, which is understood to be the fastest passenger steamer afloat, having her boilers fitted with the Howden system. About 2,000 steamers in all have now been fitted with an indicated horsepower of considerably over 6,000,000. Warships are also now very largely fitted with

the Howden system of controlled combustion; in short, practically every ship of any importance now launched is fitted with this system, which gives a greatly increased efficiency of the boiler installation, thereby reducing correspondingly the weight and space occupied. The reduction in coal consumption effected admits, in the case of merchant ships, of increased cargo space being secured through the reduction of bunker space necessary. The system also renders the propulsive power of the ship entirely independent of the state of the atmosphere. The combustion is effectively regulated quite independently of what may be the direction of the wind or the state of the temperature. This latter advantage is, obviously, a most valuable one for ships trading in the tropics. By this system also, the stokeholds in vessels so employed are rendered quite cool, and easy to work in. A considerable portion of the forced-draft apparatus for vessels so equipped, including the air-heating plant, is constructed by the contracting engineers, from complete working drawings supplied, but the more vital part of the installation, such as the fans and fan engines, furnace fronts, etc., are supplied, in every case, by the inventors and patentees themselves, Messrs. James Howden & Co., Scotland street, Glasgow.—"The Marine Review."

EXPERIMENTS WITH SUPERHEATED STEAM.

The Bavarian Revisions-Verein has lately made a large number of experiments with superheated steam to determine the effect of pressure, jacketing, intermediate superheating and number of cylinders. The full data are not yet worked out, but preliminary statements concerning them by Mr. C. Berner in the "Zeitschrift" of the Society of German Engineers give the following information:

As regards the transmission of heat through conducting surfaces, with steam velocities from 15 to 100 m.-sec. no noticeable effect on the transmission was observable. The higher velocity only produces a greater loss of pressure; thus

the resistance to flow in a coiled superheater tube is at least 20 per cent. greater than in a smooth pipe.

A peculiarity which had previously been observed, the great increase of heat transmission with increase of load on the superheater, was again noticed. This increase is usually attributed to the influence of the velocity of the flue gases and of the steam; but this view is not tenable. Only the dependence on the temperature gradient remains. This latter is, in many cases, of by far the most importance, the effect of velocity being generally negligible. In the transmission of superheated steam the tests have shown that Péclet's results, instead of being 25 per cent. too small, are in reality 70 to 80 per cent. too low. As regards the specific of superheated steam, Linde's results have been confirmed, so that Lorenz' equation gives mean results at least 20 to 25 per cent. too high.

According to the height of the steam temperature the pipe-wall temperature can lie either above or below the saturation temperature of the steam. It was found that condensation increases rapidly from the instant the pipe-wall temperature falls below the saturation temperature, but no condensation can take place if the wall is above this temperature. The limiting temperature depends on the pressure and the heat insulation. All experiments have shown that the only determining factor for the heat loss is the temperature gradient between steam and air, the loss not increasing proportionately with, but faster than, the gradient. The absolute value of the temperature of the steam is immaterial. From this it follows that the heat loss must always be greater with superheated than with saturated steam of the same pressure. The influence of uncovered parts in the piping is shown to be very great. The resistance to flow of the steam in pipes with different pressures and velocities is found to be practically the same for both saturated and superheated steam.—"The Engineering Record."

DOUBLE-ENDED WATER-TUBE BOILER.

Hitherto, the only type of water-tube boiler made by Yarrow and Co., Limited, of Poplar, has been that in which the

boiler is fired from one end only, the furnace having at the back, a firebrick wall and the necessary double casings and asbestos to protect it. But quite recently the firm has undertaken the construction of a double-ended boiler, the designs being in all other respects based upon the firm's well-known principle. The double-ended boiler has been constructed to take the place of the locomotive boiler in the torpedo boat *Halcon,* built by Yarrow and Co., Limited, for the Spanish Government nineteen years ago. The locomotive boiler seems to have done well, but it was decided by the Spanish authorities not to reproduce it but to provide one of the water-tube type, so as to conform to modern requirements. It is expected that at least a knot more speed will be obtained by this change, partly due to the saving in weight, which amounts to four tons, and partly due to a greater supply of steam. The heating surface of the new boiler is 3,450 square feet, and the grate surface 60 square feet. The locomotive boiler which it is replacing had 2,300 square feet of heating surface, and 46 square feet of grate surface. Mr. Yarrow gives these reasons for the innovation in this particular instance, but we understand that it is by no means the firm's intention to discard the single-ended type. Some of the arguments given in favor of the double-ended boiler are that it requires much less floor space for the same power than does the ordinary type, that the weight, which is always an item of considerable importance, is considerably less, and, lastly, that the constant repair incidental to firebrick surfaces is avoided. It follows from the foregoing that units can be increased in size, and the arrangement of all the steam and feed pipes is simplified.

Where a group of two or three double-ended boilers, with a single stokehold at each end, is placed in one compartment, the uptakes and funnel are not so complicated as compared with the single-ended type, having the boiler compartment with two rows of boilers and a double stokehold between. The objection is sometimes raised to the double-ended boiler that when there is any difference of air presure between the one stokehold and the other, flame may issue out of one of the fire doors when

opened into the stokehold with the lower pressure. This consideration is in some cases important, but we are given to understand that it has been amply provided for by the arrangement which Messrs. Yarrow adopt.

The advantages of the new boiler over the old will, it is thought, be sufficient to insure its adoption when possible. In fact, Mr. Yarrow is of opinion that this form of boiler will become the most popular on account of its simplicity, and the greater accessibility of all the parts. Its adoption in large installations will, it is said, cause a reduction of 12 to 15 per cent. in the weight of the boilers for a given power.—"The Engineer."

SHIPS.

UNITED STATES.

California — *Dock Trial.*—On January 30 a successful dock trial of this vessel was held at the works of the builders, the Union Iron Works, San Francisco, Cal. The *California* is one of the *Pennsylvania* class of armored cruisers, a full description of which will be found in Vol. XVII of the JOURNAL, pp. 1-41.

Charleston — *Final Trial.*—This vessel underwent her final trial January 22, 23, and is now ready for final acceptance by the Government. A full description of the *Charleston* and her official preliminary trials appeared in the JOURNAL, Vol. XVII, 754-805.

Paducah—*Final Trials.*—Satisfactory final trials of the *Paducah* were held December 12, 1905, and that vessel has been finally accepted. For description of vessel and official preliminary trials see JOURNAL, Vol. XVII, pp. 589, 1085.

Milwaukee—*Dock Trials.*—Satisfactory dock trials of this vessel were held at the works of contractors, the Union Iron Works, San Francisco, Cal., February 16-19. The *Milwaukee* is a protected cruiser of the *Charleston* class.

New Jersey—*Dock Trials.*—This vessel underwent satisfactory dock trials at the contractors' works, the Fore River Shipbuilding Company, Quincy, Mass., February 15, and will be ready for her official trials the latter part of March. The *New Jersey* is a sister ship to the *Rhode Island,* a description of which will be found in the first part of this number.

St. Louis.—This vessel is a protected cruiser of the *Charleston* class. She is now ready for the builders' trials, preliminary to the official trials, which will take place in April.

Idaho—*Launch of.*—This vessel was successfully launched at the Wm. Cramp & Sons Ship and Engine Building Works, December 9, 1905.

The *Idaho* is a battleship of the first class, with dimentions as follows: Length on water line, 375 feet; length over all, 382 feet; extreme beam, 77 feet; trial displacement on a draught of 24.8, 13,000 tons. Her motive power will be furnished by twin-screw triple-expansion engines of a collective indicated horsepower of 10,000, steam being supplied to the cylinders by eight water-tube boilers set in four watertight compartments. Her contract calls for a speed of seventeen knots. She will have one military mast and two funnels, and her estimated bunker capacity is 1,750 tons. The main battery of the *Idaho* consists of four 12-inch breech-loading rifles, mounted in two turrets; eight 8-inch breech-loading rifles, mounted in four turrets; eight 7-inch breech-loaders, placed behind casement armor, and two 21-inch submerged torpedo tubes. Secondary battery—Twelve 3-inch, six 6-pounders, two 1-pounder automatics, two 1-pounder rapid-fire guns; two 3-inch field pieces, two machine and six automatic guns.

The armor consists of a waterline belt having a thickness amidships of 9 inches, with thinner armor above the belt, forming a casemate. The barbettes for the 12-inch guns have a thickness of 10 inches on the face and 7½ inches in the rear, with corresponding turrets 12 inches and 8 inches in thickness. For the 8-inch guns the barbettes have a thickness of 6 and 4 inches respectively, and turrets 6½ and 6 inches. The protective deck has a thickness of 3 inches on the slopes and 1½ inches on the flat. The complement consists of 750 officers and men.

The *Idaho,* while not so large as some other battleships, nevertheless represents a type of the most powerful and up-to-date ship of war. She is a conspicuous illustration of the important factor electricity has become in the operation of a modern warship. Except for windlass and steering gear, practically all her other auxiliary machinery will be run by electricity. Electricity, too, furnishes illumination for the

vessel at night; there will be powerful electric searchlights and to her two auxiliary masts will be fitted apparatus for wireless telegraphy. In fact, she is to be equipped with every comfort and convenience known to modern naval construction.

Turbine Battleships. — The invitation for bids for the construction of the two new battleships authorized at the last session of Congress will permit bidders to send in, as usual, their own designs, while asking for proposals on the departmental plans. In the class of bids based on the private designs there will be opportunity for offering to build one of the battleships with turbine machinery instead of reciprocating engines. This decision has practically been reached by the Board of Construction as the result of a thorough investigation of the subject, and the observation of practical tests, including those recently conducted at South Brooklyn and in course of trips on turbine transatlantic liners. This is a notable departure from the usual designs for naval machinery, and marks a distinct step in the advancement of the mechanism of naval vessels. It is possible that the speed of the different classes of ships will, in time, be materially increased by this new mechanism, although the naval engineers are not claiming too much in any direction, and are not certain that the turbine system will furnish what is known as economical speed.

United States Revenue Cutter, No. 15.—An act of Congress approved January 12, 1905, authorized the construction of a steam revenue cutter for service in the waters of Albermarle and Pamlico Sounds, on the Carolina coast, at a cost not to exceed $175,000. The waters in these sounds being very shoal, the draught of the vessel would necessarily have to be small, and a board of officers of the Revenue Cutter Service was appointed to consider the subject, and submit a report with recommendations as to the proper dimensions for a vessel suitable for the duty required.

From the report of this Board the lines for the hull of a twin-screw, light-draught steel vessel were drawn, and a model was constructed and "pulled" in the tank at the Washington Navy Yard. From the data of these tank trials, the requisite

horsepower was decided upon, and plans and specifications for the construction of the hull and machinery were drawn up, printed and issued in January, 1906.

The vessel, to be known until launched as *Revenue Cutter No. 15*, is to be of steel, of about 408 tons displacement, with one mast, one funnel, a flush main deck, an interrupted berth deck, and a deck house with a top-gallant forecastle extending aft to the forward end of the deck house.

Length over all, feet	158
between perpendiculars, feet	148
Depth from base line at side, amidships, feet	10
Spring of beams in 30 feet, inches	7½
Breadth of beam, molded, feet	30
Draught of vessel with 25 tons of coal and 2,400 gallons of fresh water, stores and provisions, ready for sea, feet	5
Displacement at 5 feet draught, tons	408

The hull is divided into six compartmnts by five water-tight bulkheads. On the top-gallant forecastle is located the steam windlass engine, anchor davits, and a 6-pounder rapid-fire gun. From this deck a hatchway and ladder afford access to the main deck. The top-gallant forecastle deck extends aft to the top of the deck house, thus forming a continuous deck, affording protection to the main deck forward. The pilot house and chart room are located on the forward end of the deck house, the top of the pilot house being extended forward and out to the sides of the vessel to form the bridge. On the bridge and in the pilot house are installed the steering wheels, engine room telegraphs and bell pulls, binnacles, etc., and on the forward end of the bridge is a searchlight, or projector, and the telephotos keyboard. On the main deck under the top-gallant forecastle, beginning forward, are located a paint locker, shower baths for the crew, carpenter's work bench, waterclosets for the warrant officers and crew, and the refrigerators for the vessel. A hatch and ladder afford access to the berth beck. In the deck house are the following rooms: A lamp locker, three storerooms, the galley, the armory, the upper fireroom space, the engine room, the wardroom entrances, and the cabin of the commanding officers.

On the berth deck forward is the prison, the crew's space, hatches to the main hold, and rooms for the warrant officers; the berth deck is interrupted by the fireroom and engine compartments, and aft of the engine-room bulkhead continues as the wardroom deck. In the wardroom space is the wardroom country, an office, a pantry, six staterooms and a lavatory. Aft of the wardroom lavatory is the quadrant room. Below the berth deck, beginning forward, is a trimming tank, chain lockers, the main hold, and the fresh-water tanks of 1,200 gallons capacity. Below the wardroom deck are two water tanks of 2,000 gallons capacity each, and an ordnance room; and aft of this space is a storeroom for small stores. On each side of the fireroom compartment, extending up to the main deck, are the coal bunkers, of about 70 tons capacity. A water tank of 2,500 gallons capacity is worked into the after end of each bunker.

Machinery.—There are two triple-expansion propelling engines of the vertical, inverted-cylinder, direct-acting type, each having one high-pressure cylinder, 8 inches, one medium-pressure cylinder, 13¼ inches, and one low-pressure cylinder, 22 inches in diameter, the stroke of all pistons being 18 inches. The I.H.P. of these engines will be about 600 when making 220 revolutions per minute. The main valves are of the piston type for each engine, all worked by Stephenson link motion with double-bar links. Each main piston has one piston rod with a cross head working on a bar guide. The framing of the engines is of the open type, each engine having ten finished wrought-steel columns, with tie rods. The engine bedplates are of cast iron, supported on foundations built up from the frames of the vessel. The crank, line and propeller shafts are forged of steel, and are solid, except the crank pins, which have a two-inch hole bored through them. There is one condenser, of the cylindrical type, placed athwartship in the after part of the engine room. It has 858 seamless-drawn composition tubes, ⅝ inch diameter, giving a cooling surface of 840 square feet. There is one vertical, single-acting air pump, 13 inches in diameter and of 7 inches stroke, attached

to each engine, and worked by beams from the low-pressure engine crossheads.

There are two single-acting feed pumps, 1¾ inches diameter and 7 inches stroke, attached to the port engine, and two single-acting bilge pumps, 1¾ inches diameter and 7 inches stroke, attached to the starboard engine, all worked by arms from the air-pump rock shafts. All the pumps are located in the engine-room compartment, and are as follows: One independent circulating pump, of the centrifugal type, for the main condenser, fitted to take water from the sea or the bilges, and delivered through the condenser or outboard.

One independent main feed pump fitted to take water from the filter and feed tanks and discharge to the main feed pipe, and one independent auxiliary feed pump fitted to take water from the filter and feed tanks, the bilges, the sea, from a suction hose, or the condenser, and discharge into the auxiliary feed pipe, the fire main, overboard, to the salt-water side of condenser, or to the feed tanks. Two injectors to feed the boilers are located in the engine room.

One independent bilge pump fitted to take water from the bilges or the sea, and to discharge overboard or into the flushing system. One air-compressing pump to maintain a pressure in the pressure tank for the water used in the various staterooms and lavatories. One evaporator feed pump, and one distiller circulating pump, fitted to take water from the sea and discharge through the distiller, into the fire main, or the flushing system. A steam-steering engine is located in the after part of the upper engine room, where it will be readily accessible at all times.

There is one water-tube boiler, about 9 feet 3 inches long, 11 feet 9 inches wide, and 11 feet high, measured over the casing. This boiler has a total grate surface of 75 square feet, a heating surface of 3,105 square feet, and is designed for steam pressure of 200 pounds per square inch.

There is one single-ended auxiliary or donkey boiler of the horizontal return fire-tube Scotch type, designed for steam pressure of 200 pounds per square inch. The shell is 5 feet 4 inches diameter, and 7 feet 3 inches long, with one corrugated suspension furnace, 30 inches diameter, and 5 feet 10 inches long. This gives a grate area of 12.5 square feet, and a heating surface of 296 square feet. The pipe connections between the main boiler and the auxiliary boiler are so arranged that the auxiliary boiler can be cut into line with the main boiler, and used for steaming purposes, if so desired. There is one smoke pipe for these two boilers, 4 feet 3 inches in diameter, and 48 feet high.

There is one evaporating and distilling plant, located in the lower engine room, having a capacity of 1,700 gallons of potable water per twenty-four hours. This evaporator is also piped so as to be used for a feed make-up if so desired. The vessel is lighted throughout by electricity, the current being furnished by one steam turbine-driven generating set of 20 kw. capacity, with 125-volt pressure at the terminals. The wiring is of two-wire feeder system, run in steel enameled conduits in the engine and fireroom compartments and in approved molding in all other parts of the vessel. There are two 5-ampère enclosed arc lamps in the engine room and fireroom, and 120 incandescent lamps throughout the vessel. There is one night signaling apparatus or telephotos, the lanterns being suspended from an outrigger on the mast, and the combinations being controlled by a keyboard on the bridge. An 18-inch searchlight or projector is located on the forward part of the bridge. A complete outfit of hand portable lights, cargo clusters, and $\frac{1}{12}$ horsepower fans, is also installed.

There is one steam turbine-driven blower or fan installed in the upper fireroom for forced draft, the air being supplied to the ashpits of the main boiler. This fan is also connected by ducts to the various compartments of the vessel, being arranged so as to ventilate these compartments either by forcing air to them, or exhausting air therefrom.

Name of vessel.....*U. S. Revenue Cutter, No. 15.*
I.H.P., tons..... 600
Displacement, tons.....
Type of engines.....Triple-expansion, direct-acting.
Number of engines..... 2
Diameter of cylinders, inches.....8, 13½, 22
Stroke of cylinders, inches..... 18
Valve gear..... ..Stephenson link.
Connecting rod, between centers, inches..... 42

	Scale of valve circles—Double size.					
	H.P.		I.P.		L.P.	
Eccentricity, inches	1¼		1¼		1¼	
Travel of valve, inches	3¼		3¼		3¼	
Piston valves, number	1		1		2	
Side of valve on which steam is taken	Outside		Outside		Outside	
	Top	*Bottom*	*Top*	*Bottom*	*Top*	*Bottom*
Diameter of valves, inches	4½	4½	6½	6½	7½	7½
Steam lap, inch	¾	¾	¾	¾	¾	¾
Exhaust lap, inch	5/32	¼	1/16	5/32	1/16	¼
Angular advance, degrees and min.	32°20	32°20	35°20	35°20	36°25	36°25
Steam lead ang., deg. and min.	9°0	12°30	9°0	12°30	7°45	13°45
linear, inches	3/16	3/16	3/16	5/16	7/32	11/32
Cut off, in inches	13 1/16	13½	13 1/16	13½	13¼	12 15/16
decimal of stroke	.773	.729	.773	.729	.764	.718
Exhaust release, in inches	16 7/16	16 7/16	16½	16 9/16	16¼	16¼
dec. of stroke	.912	.915	.917	.922	.931	.931
Compression, in inches	1¼	1¼	1¼	1¼	2 3/16	2⅛
decimal of stroke	.091	.092	.097	.098	.121	.131
Steam opening, inch	7/8	1	⅞	1	¾	1
Exhaust opening	Full	Full	Full	Full	Full	Full
Velocity of steam in feet per second at 660 feet and 220 revolutions per minute	59.4	51.7	102.1	89	131	114
Velocity of exhaust in feet per second at 660 feet and 220 revolutions per minute	46		79		102	
Velocity of exhaust to condenser			95			

Displacement in salt water to 5-foot water line, tons	408
Total coal capacity, tons	47
Normal coal supply, tons	25
Total fresh-water, gallons	8,200
Normal fresh-water supply, gallons	2,400
Area of midship section to the 5-foot water line, square feet	143.54
immersed longitudinal section, to the 5-foot water line, square feet	693.46
the 5-foot water plane, square feet	3,345
Wetted surface to the 5-foot water line, square feet	4,366
Tons per inch immersion, 5-foot water line	7.96
Coefficient of the midship section	.937
5-foot L. W. L.	.735
displacement	.644
Frame spaced, inches	24
Section spaced, feet	10
Water line spaced, inches	12

Degree of Completion of Vessels in the U. S. Navy.

No.	Speed.	Name.	Builders.	PER CENT. completed Jan. 1.	Feb. 1.
			BATTLESHIPS.		
13,	19 knots,	*Virginia*.....	Newport News Co.....	98.	98.6
14,	19 knots,	*Nebraska*.....	Moran Bros.....	84.24	86.17
15,	19 knots,	*Georgia*.....	Bath Iron Works.....	91.24	93.35
16,	19 knots,	*New Jersey*.....	Fore River S. B. Co.....	95.5	96.5
17,	19 knots,	*Rhode Island*.....	Fore River S. B. Co.....	99.	99.5
18,	18 knots,	*Connecticut*.....	Navy Yard, New York.....	93.69	95.16
19,	18 knots,	*Louisiana*.....	Newport News Co.....	94.17	95.24
20,	18 knots,	*Vermont*.....	Fore River S. B. Co.....	70.7	73.5
21,	18 knots,	*Kansas*.....	New York S. B. Co.....	69.8	72.5
22,	18 knots,	*Minnesota*.....	Newport News Co.....	80.33	82.08
23,	17 knots,	*Mississippi*.....	William Cramp & Sons.....	47.32	49.08
24,	17 knots,	*Idaho*.....	William Cramp & Sons.....	45.39	47.19
25,	18 knots,	*New Hampshire*..	New York S. B. Co.....	33.4	37.1
			ARMORED CRUISERS.		
6,	22 knots,	*California*.....	Union Iron Works.....	87.5	88.7
9,	22 knots,	*South Dakota*.....	Union Iron Works.....	86.2	87.3
10,	22 knots,	*Tennessee*.....	William Cramp & Sons.....	91.66	94.53
11,	22 knots,	*Washington*.....	New York S. B. Co.....	91.9	94.4
12,	22 knots,	*North Carolina*...	Newport News Co.....	30.74	34.48
13,	22 knots,	*Montana*.....	Newport News Co.....	25.84	30.11
			PROTECTED CRUISERS.		
20,	22 knots,	*St. Louis*.....	Neafie & Levy Co.....	86.	88.34
21,	22 knots,	*Milwaukee*.....	Union Iron Works.....	87.9	89.8
			TRAINING SHIPS.		
1,	Sails,	*Cumberland*.....	Navy Yard, Mare Island.....	95.	95.
2,	Sails,	*Intrepid*.....	Navy Yard, Mare Island.....	97.5	97.5
			SCOUT CRUISERS.		
1,		*Chester* (turbine).....	Bath Iron Works.....	15.53	18.87
2.		*Birmingham*.	Fore River S. B. Co.....	18.2	21.
3,		*Salem* (turbine).....	Fore River S. B. Co.....	17.6	21.1
			SUBMARINE TORPEDO BOATS.		
9,		*Octopus*.....	Fore River S. B. Co.....	46.9	51.
10,		*Viper*.....	Fore River S. B. Co.....	37.8	43.3
11,		*Cuttlefish*.....	Fore River S. B. Co.....	39.9	43.5
12,		*Tarantula*.....	Fore River S. B. Co.....	38.7	43.1

AUSTRIA.

Erzherzog Friedrich. — The official steam trials of the Austrian battleship *Erzherzog Friedrich* have just been completed, with the following results: Revolutions of engines, 138; indicated horsepower, 18,340; speed, 20.57 knots, speed contracted for, 19.25 knots; power contracted for 14,000 horsepower. The coal consumption varied from 1.55 pounds per horsepower per hour to 1.85 pounds per horsepower per hour.

New Austrian Armored Cruiser. — A handsome and powerful new armored cruiser named *St. Georg*, has just been added to the navy of Austria-Hungary. The vessel was built at Pola, and is of steel, with twin screws, 383 feet between perpendiculars and 402 feet 6 inches over all, and 62 feet beam. On a draught of water of 21 feet 4 inches, the displacement is 7,300 tons. The *St. Georg* has two sets of triple-expansion engines, of a total indicated horsepower of 13,000, giving the vessel a speed of 21 knots. The boilers are of the Yarrow water-tube variety. The coal capacity is 820 tons, giving a steaming radius of 2,100 knots. The armor consists of a Kruppized steel belt 8.2 inches maximum thickness. The turrets are of 8.2-inch steel, the casemates of 5-inch, the bulkhead of 7.6-inch, and the protective deck of 1½ inch plates. The armament consists of two 9.4-inch, breech-loading guns in the fore turret, five 7.5-inch quick-firers located as follows; one in turret aft and two as bow and two as stern chasers, four 5.9-inch quick-firers, two light guns, 25 small quick-firers and machine guns, and two submerged torpedo tubes.

The navy of Austria-Hungary now consists of the following vessels:

Eighteen battleships ranging in size from 3,550 to 10,200 tons displacement; three armored cruisers from 5,270 to 7,300 tons, of which the *St. Georg* is the largest; five cruisers, 2,300 to 4,064 tons; six river monitors, 310 to 448 tons; three torpedo cruisers, 1,530 to 1,675 tons; eleven torpedo gun vessels, 350 to 1,011 tons; nine torpedo-boat destroyers, each 390

tons; fifty-five torpedo boats (first-class), 78 to 200 tons; forty-five torpedo boats (2d and 3d class), 10 to 64 tons thirty-six other vessels.

BRAZIL.

Aquidaban.—The terrible fate of the Brazilian warship *Aquidaban,* which sank near Rio de Janeiro on the night of Jan. 21 1906, within three minutes of an explosion in the powder magazine, was rendered more disastrous by reason of the fact that she carried more than her usual complement of 388, and that among the passengers were a number of naval officers, government officials, and others, who were engaged in a tour of inspection of sites for the construction of a naval yard. The *Aquidaban* was built at Poplar twenty years ago, and has had a somewhat chequered career She played a conspicuous part in the revolt of the Brazilian fleet in the autumn of 1893, and in the following March was torpedoed by the *Sampaio* and sunk off Desterro in shallow water, being subsequently raised and fitted out anew at Stettin. For a time she was known as the *Vinte-quatro de Maio.* The *Aquidaban* was a steel-sheathed and coppered vessel propelled by two screws. Her length was 280 feet, beam 52 feet, maximum draught 20½ feet, and displacement 4,950 tons. The armament consisted of four 8-inch guns, four 4.7 and eighteen smaller guns, with five torpedo tubes, two submerged forward. She carried a protective belt, varying in thickness from 7 to 11 inches, while the two turrets in échelon carrying the big guns were protected with 10-inch compound armor. Steam was supplied by eight cylindrical boilers, supplying two sets of three-cylinder engines, and her speed is given as 10 knots. Her coal capacity was 800 tons.—"Page's Weekly."

ENGLAND.

The Naval Program.—The Admiralty have now definitely completed the contracts in connection with all the ships embraced in the Naval Programme for the current year. The last of the ships—three armored cruisers of the *Invincible*

class—have now been ordered from Sir W. G. Armstrong, Whitworth and Co., Limited, of Newcastle-upon-Tyne, John Brown and Co., Limited, of Clydebank, and the Fairfield Shipbuilding and Engineering Company, Limited, of Glasgow. The Navy estimates embraced a fourth cruiser of the same type, but it has been decided to defer the construction of this vessel. The cruisers will, we understand, be the most powerful that have yet been built, but, in conformity with the desire of the Admiralty, we refrain from giving any details. It may, however, be said that they will be fitted with turbine machinery, which is to be constructed under the Parsons patent, and that the speed to be attained will, as is also the case with the offensive and defensive qualities, excel that of any of the preceding vessels. These three vessels, with the *Dreadnought*, building at the Portsmouth Dockyard, make the four armored ships which the Admiralty, in their recent exposition of naval policy, indicated as the annual instalment of new work to be commenced each year. The machinery of the *Deadnought*, as has already been indicated, is also of the Parsons type, and the contract for it has been placed with Messrs. Vickers' Sons and Maxim, Limited, of Barrow-in-Furness. The other new work includes five ocean-going torpedo-boat destroyers, which, we understand, are to have a speed of 33 knots, and these have been ordered from Sir W. G. Armstrong, Whitworth, and Co.; Messrs. Cammell, Laird, and Co., of Birkenhead; Messrs. J. I. Thornycroft and Co., of Chiswick; Messrs. Yarrow and Co., of London; and Messrs. J. S. White and Co., of Cowes. Twelve coastal torpedo-boat destroyers were also included, and the contracts for these have been placed with the three last-named firms; five will be built by Messrs. White, five by Messrs. Thornycroft, and two by Messrs. Yarrow; while eleven new submarine boats are to be built by Messrs. Vickers' Sons and Maxim, Limited, of Barrow-in-Furness. This constitutes quite a satisfactory year's programme, especially when it is borne in mind that, consistent with the strong progressive policy which has characterized Sir John Fisher's administration of naval affairs, each type of vessel will far

exceed, alike in fighting power and speed, the predecessors of its respective class. Obviously the determination is that each unit of the British Navy should have a considerable superiority over a prospective combatant in any "friendly" Power; and the realization of this idea could not be in better hands than those of the First Lord and the chiefs of the Technical Department at Whitehall.—"Engineering."

Black Prince.—This new first-class armored cruiser has completed her steam trials with the following results: On the second thirty-hours' trial the engines developed 16,699 I.H.P., and the mean speed during the whole of the thirty hours was 21.6 knots. On the eight-hours' full-power trial, which was made off Plymouth on the 12th ult., the engines developed 23,939 I.H.P., giving a mean speed on the measured mile of 23.66 knots. There was a marked absence of vibration and the ship proved to be very steady in a sea-way.

The interesting feature about the trials of this vessel, apart from the splendid speed achieved, is the fact that the machinery has been built under the new standardizing regulations, with the result that almost any unit may be taken out of the *Black Prince* and will fit at once in a sister ship—into the *Duke of Edinburgh, Cochrane, Natal* or *Achilles.* The cost of this advantage is considerable, as the large number of gauges to ensure accurate diameters, of jigs to ensure that all holes on flanges and for connection by means of bolts, nuts and screws are the same, and of templates to ensure a similarity in sizes, cost about $60,000 for each ship, equal to about 5 per cent. on the price of the machinery, including boilers. The practice now regularized is for the firms who may secure the order for one of several sets of engines for ships of a class to agree upon the details of the design, and to arrange that each shall make the gauges or jigs for certain parts—cylinders, pistons, etc.—for all the engines. Thus the work of preparing the standards is divided, and as they are severally made from one pattern, etc., similarity is guaranteed, and the cost minimized. But withal the expense is heavy. It is true that great advantage accrues, as in the event of breakdown spare parts

may be more easily supplied. In the merchant service this practice has been applied for some time, more or less extensively. The *Black Prince,* on her eight-hours' full-power trial, made 23.66 knots with 23,939 indicated horsepower. The designed power was 23,500 indicated horsepower, and it was expected that this would give 22.33 knots. This gain in speed is highly creditable to the director of naval construction, for the vessels are of 13,550 tons displacement, while in the case of the *Devonshire* class the power for the same speed and a displacement of only 10,200 tons, was not much less—about 22,000 indicated horsepower. The additional displacement represents great gain to offensive and defensive qualities, the new ships, for instance, having six 9.2-inch and ten 6-inch guns, as compared with four 7.5-inch and six 6-inch guns.—"Practical Engineer," January 28, 1906.

Attentive.—In the eighth hour of her trial, September 12, H. M. S. *Attentive* obtained the following results: Steam in boilers, pounds, 250; steam at engines, pounds, 245; air pressure in stokeholds, inches, 3.1; vacuum, inches, 24.5; mean revolutions, 260.6; indicated horsepower, 16,879; speed, 26.2. This was the vessel built by Armstrong Whitworth & Co. in competition with other firms, all of whom were given a free hand in the attainment of certain desired ends instead of being limited to Admiralty designs as is usual. The other firms in competition were Messrs. Cammel Laird, the Fairfield Company, Vickers' Sons, and Maxim. The lengths of the different vessels vary between 374 feet and 360 feet. The beams between 40 feet and 38 feet 3 inches, and the draughts between 13 feet 10½ inches and 12 feet 6 inches. The displacements range between 3,060 tons and 2,620 tons, and the horsepower developed on trial between 17,500 and 14,300. The speeds in the full horsepower trials vary between 25.88 and 25.07 knots. The two vessels built by the Armstrongs, the *Attentive* and *Adventurer,* have made the best sustained speed, 25.4 and 25.88, the *Attentive* as shown above reaching 26.2 in the last hour of the trial. "This," says the "Engineer,"

"coming at the very end of an onerous trial, speaks volumes for the efficiency of her boiler equipment."

H. M. S. Encounter.—The following results were obtained at the thirty-hours' steam trial at 70 per cent. of full power, which the new cruiser *Encounter* underwent on October 14 and 15, 1905: Steam pressure at boilers, 227 pounds per square inch; steam pressure at engines, 210 pounds per square inch; vacuum, 26.0; revolutions per minute (mean), 159½; indicated horsepower per hour, 9,008; speed, 19.0 knots per hour; coal consumption per indicated horsepower per hour, 1.98 pounds. At a full-power trial on the 18th ult., the following results were obtained: Steam pressure at boilers, 230 pounds per square inch; steam pressure at engines, 218 pounds per square inch; vacuum, 25.3; revolutions per minute, 178; indicated horsepower per hour, 13,006; speed, 21.1 knots per hour; coal consumption per indicated horsepower per hour for all purposes, 2.09 pounds.

Warrior.—This new first-class cruiser was launched from Pembroke Dock on Saturday. She is 505 feet long, 73 feet 6 inches in breadth, has a mean draught of 27 feet, a displacement of 13,550 tons, an estimated horsepower of 23,500 and speed of 22.33 knots. Her total weight, including armor, is 8,015 tons. The *Warrior* belongs to the first cruiser class in the British Navy in which the 6-inch gun has been dispensed with, and, although this has involved a reduction in the number of guns, the gain is enormous in the penetrating power of each shot fired. Assuming favorable conditions, so far as rapidity of fire is concerned, the *Warrior* will develop practically 835,000 foot-tons of muzzle energy for each minute's fire as compared with 829,000 foot-tons in the preceding ships; while but four years ago our fast cruisers—those of the County class, fitted only with 6-inch guns—developed a total of only 542,000 foot-tons. The important point, however, is that the *Warrior* may fire per minute forty-three shots, which can be effective at three miles distance, as compared with twenty-four in the immediately preceding ships of the *Duke of Ed-*

inburgh class. The cost of the *Warrior* is about £1,100,000. —"Page's Weekly," Dec. 17, 1905.

Cricket.—There was a successful launch on Tuesday, January 23d from the works of Messrs. J. S. White and Co., Limited, East Cowes, Isle of Wight, of H. M. S. *Cricket,* the first of the new coastal destroyers building under the Naval Programme of 1905-6. These vessels will be propelled by turbines built by Messrs. J. S. White and Co., Limited, under license from the Parsons Marine Steam Turbine Company, Limited, the power of the machinery being about 4,000 indicated horsepower. The steam will be supplied by two boilers, each of 2,000 horsepower, of the White-Forster pattern, made by the same firm, these boilers being fired by liquid fuel, on a system which has been experimented with successfully by the British Admiralty for some two or three years. No coal stowage is provided in the vessels, and they will rely entirely on the liquid-fuel installation. The auxiliary machinery in connection with the turbines is being provided by Messrs. G. and J. Weir, Limited, of Cathcart, Glasgow, as are also the feed pumps and the liquid-fuel pumps. The fan engines and fans are supplied by Mr. Peter Brotherhood, Westminster-bridge, London, S. E., and the evaporators and distillers by Messrs. John Kirkaldy, Limited, London, E. C. The *Cricket,* which is about 230 tons displacement, is representative of a type of vessel which, with a speed of 26 knots, maintained on an eight-hours' full-power trial, is a development of the late first-class torpedo boats driven by reciprocating machinery, which, with slightly less displacement, have a speed of 25 knots with 3,000 horsepower.

Dreadnought. — Few, if any, warships built in this or any country have excited the same keen and universal interest as the *Dreadnought,* which was launched at Portsmouth Dockyard February 10th in the presence of the King. This is due to the fact that the vessel is beyond doubt, the most powerfully armed, the best protected, and the fastest battleship ever laid down. It embodies, not only in the general elements of design, but also in the details of equipment, the great experi-

ence and knowledge of strategy and tactics of Sir John Fisher, the First Sea Lord of the Admiralty, universally regarded as the leading exponent of our naval policy. His ideas have found expression in concrete form through the co-operation of Sir Philip Watts, K. C. B., the Director of Naval Construction, who has ever shown skill and initiatory courage, and these qualities have attained their fullest scope in the design of this new ship.

The armament calls for first attention, and, as we explained in a previous article—which contained the first information published regarding this ship, and practically all the information thus far made public—all of the ten guns of the primary armament are of the 12-inch caliber; indeed, in all the large armored ships which have recently been laid down in the British Navy, this is the only gun adopted. There are many advantages to be derived from unification of caliber: the supply of ammunition at the naval bases is enormously simplified, while on board ship there is less likelihood of confusion, and a greater certainty of being able to use the best guns effectively up to the last moment of an engagement, because the remaining store of ammunition may be conveyed from one magazine to another in order to serve the gun most conveniently placed. This unification is being universally accepted, partly as a consequence of British advocacy, but mostly because of the reports by attaches present at the Battle of the Sea of Japan. There is not, however, the same general agreement as to the caliber of gun which should be adopted, owing solely to the amount of attention that must be devoted to the question of cost. Most of the Powers, however, accept the 12-inch gun, although in France there is a strong advocacy of a new 11-inch weapon, while in the new Russian cruisers the 10-inch gun is being adopted. The deficiency in caliber in the latter instances is partly compensated for by higher velocities.

Recent war experience has shown that the gunner should be able to strike any target that can be properly aimed at, or is within view; the perfection of sighting apparatus is such that these are almost synonymous terms. But it is not suffi-

cient to merely strike the target—it must be destroyed; and to attain this object many naval officers contend that more than mere striking velocity is required. The mass of the shot counts, and therefore more destruction will be attained with a projectile of 850 pounds weight fired from a 12-inch gun than with a 500-pound shot from a 10-inch gun, or a 380-pound shot from a 9.2-inch gun, even although these latter, by reason of their greater velocity, develop nominally the same striking energy. This was obvious by examination of the Russian ships sunk at Port Arthur, because the smaller shots from the 6-inch guns, and even many from the 12-inch guns, failed in what, a century ago, was termed "smashing" effect, when the old carronades were introduced. Volume in shells is also of importance for the same reason, although the destructive effect of the contents is directly relative to size. There is therefore little chance of the 12-inch gun being discarded for a lighter weapon of high velocity, even although the direct saving in weight on board ship for a pair of guns and their mountings within a barbette is between 170 and 200 tons, while the indirect saving, consequent upon a possible reduction in the dimensions of the ship and the power of machinery, is nearly threefold. The probabilites are rather in favor of even greater calbers.

There is, however, no obvious reason why the velocity of the 12-inch gun should not be increased in a way corresponding with that of the new French 11-inch or the new Russian 10-inch guns. It is said that in the large Russian cruiser, which the Vickers Company are building there will be fitted 10-inch guns to attain a velocity in service of 3,000 feet per second with a normal pressure in the chamber and bore. Such a velocity has only been exceeded on purely experimental trials, and has thus far not been regarded by any means as a normal condition, so that great interest will be taken in this new weapon. Looking, however, to the success hitherto of the Vickers ordnance, and to the great influence that such success has had on warship power generally there seems no reason to expect other than a satisfactory issue from this further ad-

vance. The effect will be a development of energy in the 10-inch gun corresponding almost to that of the 40-caliber 12-inch gun; but were the same velocity applied to the 45-caliber 12-inch gun, its power would be proportionately increased. In the *Dreadnought,* however, the guns of 45 calibers are to have a velocity restricted to 2,850 feet per second, and will give a muzzle energy of 48,000 foot-tons. These guns are being manufactured, with their mountings, by Sir W. G. Armstrong, Whitworth, and Co., Limited, and Messrs. Vickers Sons and Maxim, Limited.

Up till quite recently the 12-inch gun adopted on British ships has been of only 40 calibers in length, and the velocity has been more usually 2,500 feet or 2,600 feet per second. Each gun of the *Dreadnought* will therefore be something like 30 per cent. more powerful than those fitted in ships of two years ago, double the power of the 9.2-inch gun, nearly five times the power of the 7.5-inch gun, and eight times that of the 6-inch gun. It will thus be readily understood that in offensive power the new ship is an enormous step forward. In the case of the *Royal Sovereign,* built in 1892, and with four 12-inch guns of moderate power, the collective energy at the muzzle of one round from all of the guns was only 159,-619 foot-tons. The vessels of ten years later had a collective energy of 194,400 foot-tons, and in these vessels there were only four 12-inch guns, the remainder of the armament being 6-inch weapons. In the *King Edward VII* class, four 9.2-inch guns took the place of some of the 6-inch quick-firers, and, as a consequence, the energy was increased to 270,000 foot-tons. In the first battleship designed by Sir Philip Watts when he returned to the Admiralty—the *Lord Nelson*—all of the 6-inch guns were discarded, and the armament became four 12-inch guns with ten 9.2-inch guns; and now in the case of the *Dreadnought,* with ten 12-inch guns, the collective muzzle energy of one round has gone up to 480,000 foot-tons. Thus in ten years, from 1892 to 1902, there was an increase of 22 per cent., while in the past five years the advance has been 147 per cent. More important still, every shot from the *Dread-*

nought will be effective at six miles range; whereas in the case of the ships of three years ago the maximum range was not more than five miles, and even then the number of shots fired within a given time was only 30 or 40 per cent. of that of the *Dreadnought.*

It is contended, however, that there is a tendency to make warships merely floating gun-carriages, and that calculations as to the collective energy of one round, and also of the weight of shot fired in a given time, are illusory. As regards the former, it may be accepted, so far, at least, as the *Dreadnought* is concerned, that the increase in gun power which we have notified has not been attained at the expense of any other quality which goes to make up fighting efficiency. The new ships will meet British Admiralty conditions; but there will be a more common-sense interpretation of this phrase, and less of needless duplication. This has too often been justified as an effort to ensure reliability, which, after all, is only a comparative condition. The personal equation has been frequently undervalued. As to the measurement of gun power by the standard of weight of shot and collective energy per unit of time, there is no other basis embracing caliber, length, velocity, and energy of gun, and the efficiency of the breech mechanism and mountings. It is true that the maximum rapidity of fire attainable may not be utilized in any engagement. A large measure of the success of the Japanese Fleet was consequent upon their power of restraint, and of making sure that every shot represented value in progress towards the destruction of the Russian ships. But it is necessary that the guns should be there, and that the mechanism and mountings should give a great rapidity of fire for any emergency requiring concentration; and the measure, under some common denominator, of each element constituting their power is as important, in estimating the design, as the horsepower of the machinery, which is not developed at all stages of the engagement.

The distribution of the armament on board the ship is equally important. The offensive power being concentrated in

fewer guns, it becomes necessary that each gun should be capable of use through a wide angle; in other words, that more of the guns than hitherto should be capable of firing on either broadside. The obstruction offered by deck erections can always be minimized, but there is great difficulty in arranging magazines at the base of the gun-mountings, so as to ensure an easy and continuous flow of ammunition to the guns. In a modern high-speed battleship, however, 160 feet to 200 feet of the length of the ship is required for the machinery and coal bunkers; and it is doubtful if the expedient of placing the magazines between the boiler rooms is prudent, in view of temperature and possible explosions. There is consequently a strong presumption in favor of placing the guns, whose mountings are of great depth, and whose magazines require a considerable area, forward and abaft the machinery. The Italians are said to favor the fitting of six pairs of 12-inch guns in the center line of the ship, and of placing the magazines alternately with boiler and machinery compartments. The American proposal is to fit pairs of 12-inch guns, in two barbettes at the bow and in two at the stern, all eight guns being in the center line of the ship, the inner barbettes in each case being at a higher level than those in front, so that all four guns in the forward part of the ship may fire ahead as well as on either beam, and that all four guns aft may fire astern as well as abeam. It should be stated that these guns will not be superposed, as in the case of some of the earlier American ships, but will be entirely separate. The arrangement is open to the objection that the blast from the guns in the rear pair may considerably disturb the men working the guns in front, and especially the captain at the more or less exposed firing station, while the unignited gases entering through the ventilating holes on the top of the hood may have a stupefying effect on the whole gun crew.

In the *Dreadnought* both these objections have been overcome, the arrangement which Sir John Fisher and Sir Philip Watts have devised affording the maximum of efficiency. On the forecastle there will be mounted two 12-inch guns in a

barbette, the center line being considerably above the water level. On each side a short distance to the rear, there will be two other pairs of 12-inch guns on the upper-deck level, and in order to enable these guns to fire ahead an embrasure is formed at each side of the forecastle, so that all six 12-inch guns may take part in a running fight. At the same time four of them can be used on each broadside. Aft there are two pairs of guns, both in the center line of the ship, one pair to the rear of the other; but with this difference, as compared with the American design, that both pairs of guns are on the same level and are a considerable distance apart. The four guns, therefore, cannot be fired astern, although they have a very considerable arc of training abaft and forward of the beam. Instead of placing the guns in the fore-and-aft line they might have been mounted in echelon, but the objection to this is the same as that made to the American plan. Presuming the four guns mounted in echelon to be fired on the beam, the blast and unignited gases from one pair of guns might seriously hinder the work done by the other pair. As it is, with the gun muzzles in line, and a considerable distance apart, there is less likelihood of interference the one with the other, especially as the admiral would make it a point in his tactics to secure the weather berth. The *Dreadnought* arrangement, it is true, reduces the astern fire to two guns, which is less than in any preceding ship where there are either 9.2-inch guns or 6-inch quick-firers on each quarter. But the pair of 12-inch guns should be adequate, in view of the other qualities of the *Dreadnought,* in comparison with probable combatants. It is not probable—at all events, for several years—that any of our possible enemies will build a ship with greater offensive power, or with more admirably distributed and effective armor protection. That being so, one is justified in assuming that the *Dreadnought* will not require to run before a superior ship. Moreover, her speed of 21 knots would probably enable her to outclass any more powerfully-armed vessel, as in most foreign Powers the question of cost must militate against high speed with such gunpower. The

tactics would be rather to swing the ship, so as to bring to bear upon a following enemy the whole broadside. This was undoubtedly Admiral Togo's procedure, and, as events showed, it was the correct one.

The *Dreadnought,* therefore, besides having ten 12-inch guns, will have the enormous broadside fire of eight 12-inch guns, which far excels anything in previous ships. Since the range in battleship engagements of the future is likely to be not less than five miles, and as it is essential that each shot must have a smashing effect, apart altogether from the power of penetrating armor, it may be taken that nothing less than a 10-inch gun is of any avail; so the 12-inch gun confers undoubted and necessary superiority. The *Dreadnought,* even in a broadside action, is therefore equal to any two other ships afloat. None of the guns are at a less height than the upper-deck level, and the two forward barbette guns are on the forecastle. This very desirable improvement, which has been carried out in all ships since Sir Phillip Watts went to the Admiralty, adds materially to the fighting efficiency, because, in a heavy sea, with the ship rolling through 20 degrees or 25 degrees, the guns formerly placed on the main deck could only be fired intermittently, owing to the disappearance of the target from the sighting position. In the new ships, with weapons on a higher plane, the captain of the gun need never lose the enemy on his sights.

Another important point in reference to the armament of the modern ship is the protection against attack by torpedo and submarine boats. Hitherto in the British service the 12-pounder gun has been regarded as sufficient, but the surprising success of the Japanese destroyers in their recent engagements, even taking into account the preparatory work of destruction and demoralization by the battleship and armored this type of craft has great potentialities, and that a rapid-firing weapon of great power is needed to check small craft cruiser of the previous day, has reawakened the feeling that maneuvered with such daring and courage as was shown in the Sea of Japan. The 12-pounder gun has not what we have

already termed a sufficient "smashing" effect. The Japanese in their new battleships have decided to fit a numerous battery of 4.7-inch guns firing 45-pound shots, and curiously enough the Russians in their new cruiser are adopting the same weapon, and are fitting over twenty of them. In the *Dreadnought* the intention is to adopt an entirely new weapon, using an 18-pound shot, and many may have doubts as to the adequacy of this new weapon, in view of the immense superiority of the vessel in every other respect.

The placing of the guns on the upper deck has materially simplified the arrangement of the armor, and here it may be interjected that the adoption of turbines has assisted towards this higher gun-platform, because the weights with turbine machinery are lower in the ship, and thus the center of gravity is considerably lower; at the same time the top hamper in the ship has been reduced. The main belt in the way of the machinery has been increased in thickness to 10 inches, and the upper deck is armored. The gun mechanism is protected by thick, heavy hoods, as in the case of the earlier barbette guns; and the gun mountings, while largely protected by the main broadside armor, are further shielded by armor barbettes or cylindrical casings.

The adoption of the steam turbine has not only increased the speed, but has resulted in the improvement of the maneuvering quality of the ship. Four shafts are adopted, and this has greatly facilitated the fitting of a double stern with two rudders—a form of stern advocated for some time for heavy battleships. The cutting away of the deadwood in combination with a balanced rudder has improved the turning moment of later single-stern battleships by 30 per cent.; and as the double rudder enables a larger area to be utilized effectively, without increasing the torsion on the threaded shaft of the steering gear, there will be still better facility in maneuvering. While there is no change so far as the upper works are concerned, the stern of the ship is doubled under water, with two rudders quite 20 feet apart. The deadwood is cut away in each case to provide an aperture for the propeller on each of

the inner shafts, while the two outer shafts project in the usual way beyond the shell of the ship some distance forward. This duplication of rudder also reduces the stress on the steering gear. It has been found in some ships, where the one rudder is operated from the rudder-head crosshead by rods connected through nuts to a shaft with right and left-handed screw-thread, that the torsion on the shaft is very severe when the rudder is put hard over, especially when the ship is going astern at high speed. The duplication of steering gear with two shafts will greatly reduce the stress. The steering gear, which is of the Napier Brothers type, will be of the usual design.

The contract for the turbine machinery was placed with Messrs. Vickers Sons and Maxim, Limited, and it is anticipated that with the four propellers running at over 300 revolutions, the power developed will be equal to 23,000 indicated horsepower. There will be two high-pressure turbines and two low-pressure turbines, each on separate shafts, and each shaft will also carry an astern turbine, two of which will take high-pressure and two low-pressure steam. The high-pressure main and astern turbines are to be on the wing shaft, and the two inside shafts, in addition to carrying the low-pressure ahead and astern machines, will also have turbines of small diameter for cruising purposes. Steam for the low powers will pass from the boiler into the cruising turbines, thence to the high-pressure wing turbines, and back to the low-pressure turbine before entering the condenser. This will enable a full range of expansion to be economically attained, even with a small volume of steam.

The steam pressure is to be higher than in any previous turbine ship, as the eighteen Babcock & Wilcox boilers are to be worked at 250 pounds pressure, which will be slightly reduced at the high-pressure turbines. The boilers, consistent with the latest practice, will be fitted for working not only with coal, but with oil fuel.

In order to reduce the power necessary to attain a speed of 21 knots, and reduce the draught for a given displacement—

the *Dreadnought* when ready for sea will be about 18,000 tons on 26 feet draught—it was decided to increase the length of the ship from the 410 feet of the *Lord Nelson* to close upon 500 feet, with a beam of 82 feet. This increase in length had the further advantage that it afforded greater room forward and aft for magazines under the 12-inch guns without interfering with the under-water torpedo-tube gear in connection with the five submerged tubes. The larger magazine will, of course, be forward, where there are six 12-inch guns.

All the executive officers will have their quarters forward. It is somewhat surprising that hitherto the Admiralty have continued the location of the officers in the stern of the ship—a survival of the old sailing-ship and paddle-steamer days—while in the merchant service the change was made thirty years ago. The conservatism which has hitherto been maintained in the service called for the isolation of the admiral in a flagship, and the commanding officer in other vessels, and this has been the excuse for placing the principal rooms at the stern, far removed from the venue of work. In most ships, even in ordinary service, the day cabin of the captain, which is under the bridge, is most frequently used, and in action the ordinary rooms of the officers would seldom be utilized.

In determining, therefore, to entirely re-arrange the habitable quarters of officers and crew, the Admiralty have taken a wise step. There is no reason whatever why the forecastle of the ship should not be made as attractive and as exclusive as the stern, and under ordinary steaming conditions there will be more comfort, as, even with reciprocating engines, vibration is least felt forward.

It will thus be seen that from first to last the Admiralty have brought to bear upon the design of the *Dreadnought* not only a well-directed experience, but a common-sense determination to retain only those traditions of Admiralty practice which improve the fighting efficiency of the ship, and not those influences and traditions which have accumulated in the hands of successive Boards, who, not having the courage to prefer the modern to the old, compromised matters by adopting both.

One other feature which deserves reference is the rapidity with which the ship has been constructed. Nominally, the keel was laid at the beginning of October, but a large quantity of preparatory work had been done before this.

Even so, however, it may be said that from the time when instructions were given—about seven months ago—there has been built into the ship 5,500 tons of material, and it is anticipated that the progress of the machinery, guns, gun mountings, and armor will enable the vessel to be completed and be on her trials in the early weeks of 1907. The completion, therefore, of such a vessel within eighteen months—apart altogether from the fact that a special effort has been made, and that other work may have been delayed or somewhat neglected—is a triumph of British shipbuilding which cannot be excelled, if, indeed, it can be approached, in any other country.—"Engineering."

Petrol Boats and Steam Boilers at Poplar.—On Jan. 19 steam trials were run in the Thames, between Blackwall and Greenwich, of a small torpedo boat constructed by Messrs. Yarrow & Co., at their works at Poplar. This boat is 60 feet long and 9 feet wide, and is constructed of steel, being strongly built, so as to be capable of being lifted. The design provides for a torpedo tube aft and a machine gun forward. The point of interest about the boat is her propelling machinery, which consists of three sets of internal-combustion engines, each of which drives its own screw. The two wing propellers are driven by engines of about 120 horsepower, whilst the center propeller has a 60-horsepower engine. The central engine only can be reversed, this being sufficient for all practical purposes. These engines have been constructed from designs by Mr. Napier, and work on the usual Otto cycle. No endeavor was made on Friday to take any accurate records of the speed of the boat, but on trials that have been made for the purpose it has been found that she will run at a speed of 25½ knots, although we understand this speed has been considerably exceeded on other occasions. Messrs. Yarrow state that the best speed that could be obtained under the same conditions

with a steam-driven vessel would be 20 knots, thus giving an advance of 5½ knots in favor of the internal-combustion engine.

A good deal of advantage, however, is attributed to the particular kind of model which can be adopted with the light internal-combustion machinery. The saving in weight due to using petrol engines is put down at 50 per cent., and this lightness enables the boat to be formed so that when the engines are exerting their full power the hull rises on the surface of the water, and, as it were, slides along the top. On January 19, when the boat was run with her full power, and was doubtless traveling at the maximum speed of 25 to 25½ knots, the effect was very evident to those on board or watching the progress of the vessel from another craft. Beyond the feather of loose water at each bow, which, however, made a great show for a small volume, there was very little wave-making disturbance. The bottom of the boat is quite flat, almost like a barge, and there is no sharp forefoot. The weight of a boat of this kind, including hull and propelling machinery, does not exceed 8 tons, and it would therefore be easy to lift such craft on board other vessels, whilst for coastal defence these little swift vessels would be very efficient, it being stated that the cost of one modern destroyer would be equal to that of about fifteen of these petrol-driven torpedo launches. The smaller target is a matter of considerable military importance. Beyond this the engine-room staff in these vessels is about half that which would be needed for steam driving, so that the loss of life would be less if one of these boats were sunk. The danger of using petrol is the chief objection that has been raised by naval authorities against the adoption of the internal-combustion engine. In order to minimize this, the petrol tank is placed on the aft part of the hull, and is quite separate from the hull structure, so that in case of damage to the tank the petrol would flow into the sea and not into the boat. The storage is aft, so that in case of the petrol taking fire when the boat was traveling, the flames would pass away from the hull.

We are not aware whether the boat has been built to the order of any government.

The torpedo tube and machine gun were not in place on the trial; but we should think, considering the speed at which the boat traveled, and the good sea-going qualities obtained by her light machinery and wide hull, that most enterprizing governments would soon adopt craft of this nature, taking the risks that have to be run through the use of petrol—risks which could be minimized by ordinary precautions.

It may be stated that Messrs. Yarrow & Co. have in course of construction for the British Navy two coastal destroyers, the *Mayfly* and the *Moth*. Also two destroyers for the Greek Navy, which are to be built on the same lines as those constructed by the firm for the Japanese Navy. There are also in course of construction, for the Government of Uruguay, five shallow-draught vessels, 85 feet long and 16 feet beam, the draught being only 12 inches with all machinery on board and steam up. There are also two small vessels for India. All these shallow-draught vessels are propelled by twin screws working in funnels fitted with a Yarrow hinged flap. This arrangement enables an additional mile an hour to be obtained.

An interesting feature in the boiler shop was a double-ended Yarrow boiler which has been constructed for the Spanish Navy. There is a common furnace, firing being carried on at each end. This does away with the fire-brick wall at the end, the latter a source of considerable trouble, as it requires frequent repair, besides which it gives additional weight. The increased size of the boiler obtained in this way enables the steam and feed-pipe system to be simplified. By using one double-ended boiler in place of two single-ended boilers, a reduction in weight of about 12 per cent. is obtained. Recently nineteen Yarrow boilers have been despatched to Pembroke for the new *Warrior,* whilst there are in hand twenty-four Yarrow boilers for H. M. S. *Shannon,* which is to have engines of 27,000 horsepower. In the engineering department a number of steam turbines of the Parsons type are in course of construction.—"Engineering."

FRANCE.

French Armored Cruiser. — The *Dupetit-Thouars,* a new French armored cruiser of 9,516 tons, 19,600 indicated horsepower, built at the Toulon Naval Yard, has recently completed her steam trials. The vessel is 452 feet 9 inches length on deck, 460 feet over all, 63 feet 8 inches beam, and draws 24 feet 6 inches of water.

Her machinery, supplied by Schneider & Co., of Creusot, consists of three independent, vertical, three-cylinder, triple-expansion engines of equal power, each working a propeller. These engines are supplied with steam from twenty-eight Belleville boilers, having a total of 113 square meters (1,216 square feet) grate area; a total heating surface of 3,580.04 square meters (38,540 square feet), and are subjected to a test pressure of 20 kilogrammes (294 pounds per square inch).

The four series of trials made were the following:

1. *Coal Consumption on Trial at 1,800 Horsepower.*—According to the terms of the contract this trial was to extend over six hours, the coal consumption to be from 650 to 700 grammes (1.4 pounds to 1.5 pounds) per horsepower-hour.

The actual consumption during the trial amounted to but 519 grammes (1.1 pounds) per horsepower-hour, thus showing a very satisfactory saving, the result of which will be to increase in a marked degree the radius of action of the ship.

2. *Coal-Consumption Trial, at 14,000 Horsepower.*—This was carried out on January 6, and lasted six hours. The consumption contracted for was 750 to 800 grammes (1.6 to 1.7 pounds) per horsepower-hour.

The actual consumption only amounted to 559 grammes (1.2 pounds per horsepower-hour.)

3. *Maximum-Power Trial.*—This was delayed for about one month owing to slight damage sustained by the hull, the ship, under the action of a high wind, having become foul of the dock wing walls. The port-side false keel was also torn by coming in contact with a mooring buoy. These damages were made good, and the trials were resumed on February 14.

The conditions to be fulfilled on this occasion were the following: Duration, three hours. Power to be obtained 19,600 horsepower. Combustion per square meter of grate area to be 160 kilogrammes (32.8 pounds per square foot) per hour.

The results obtained largely exceeded all expectations. The power developed reached 22,000 horsepower, while the rate of combustion did not exceed 140 kilogrammes per square meter of grate area (28.6 pounds per square foot) per hour.

The runs made on the new measured mile at the Hyéres island in the Mediterranean showed an average speed of 22.05 knots, instead of the 21 knots contracted for.

4. *Trial at Normal Power (10,000 Horsepower).*—A recent naval order specifies that this trial, which lasts over twenty-four hours, had to take place not later than three days after the maximum power trial. The coal consumption contracted for was 750 to 800 grammes (1.6 pounds to 1.7 pounds) per horsepower-hour. The actual results obtained were no less favorable than those of the preceding trials. The coal consumption amounted to but 590 grammes (1.3 pounds per horsepower-hour.

The admiral in charge of the trials expressed himself most highly satisfied with the performance of the ship. The results obtained and recorded above reflect the highest credit upon Schneider & Co., and the Delaunay-Belleville Co.

The vessel has a 6-inch Harveyized nickel-steel belt, ending a short distance from the stern in a 34-inch transverse bulkhead. The upper belt is 3.7 to 2.4 inches, but does not seem quite up to the maindeck casemates. The protection deck is 2 inches in thickness. The turrets are of 6-inch steel, with 4-inch bases. She carries two 7.6-inch breech-loading guns in turrets forward and aft; eight 6.4-inch quick-firing guns in casemates (unprotected); four 3.9-inch quick-firers in semi-circular shields; sixteen 3-pdr. and six 1-pdr. quick firers or revolving guns, and two submerged torpedo tubes. Her complement is 540 men; fitted as flagship she will carry 583 men. Sister ships are the *Admiral Gueydon* and the *Montcalm*.—"The Nautical Gazette."

The Armament contemplated for the three battleships to be laid down in 1906, which form the first instalment of the new program of construction to be shortly commenced for the French Navy, consists of four 12-inch guns, twelve 9.6-inch, sixteen 3-inch, eight 1.8-inch and two torpedo tubes.

French Torpedo Boats. — In the French Navy there are six "torpedo cruisers," fourteen "contre torpilleurs," forty torpedo-boat destroyers and forty-three sea-going torpedo boats, all of which have twin screws; two hundred single-screw torpedo boats of the second class (sixty of which are under construction) and about fifty of the second class, all with single screws, as well as fourteen of a smaller type known as vedette boats. A typical destroyer is the *Pertuisane*. This vessel, while not one of the fastest, is one of the largest of the fleet, being 184 feet long and 20.6 feet beam and of 10 feet maximum draught, at a displacement of 303 tons. The indicated horsepower is 5,700, which is rated to drive the vessel at 27 knots, although on trial a speed of 1.5 to 2.5 knots in excess of the requirement was made on vessels of her class. The *Pertuisane* carries two 14-inch torpedo tubes, one 9-pounder and six 3-pounder quick-firing guns.

France is now building fourteen destroyers and sixty torpedo boats. An interesting feature of construction is the attempt to place an armored belt around the larger vessels. Some of the new French destroyers are, it is said, to be almost completely protected along the waterline with 2-inch hardened-steel belts. These belts will render them invulnerable to 6-pounder and 3-pounder guns, and tolerably safe against 12-pounder, except at comparatively short range.—"The Nautical Gazette."

GERMANY.

The German Cruiser Ersatz Wacht is to be built at the Vulcan yard, Stettin, and will be supplied with turbines. These will be provided by the German Parson's Turbine Company, and the "Marine Rundschau" states that they will provide for 13,600 horsepower, and for a speed of 24 knots. The displacement will be 3,410 tons, as compared with 3,250 tons

in the case of the *Lubeck*, which has a speed of 23 knots. Turbines are also to be supplied to one of the series of destroyers *G. 132* to *G. 137*, which are to be built in the Germania yard, Kiel.

Torpedo Boat S 126 collided on November 17 near Buelk with the small cruiser *Undine*. The torpedo boat sank and one officer and thirty-two seamen were drowned. The disaster occurred during the maneuvers in Kiel Bay. A torpedo-boat division was making a regulation attack upon the *Undine*, which had blinded her lights. Later she suddenly used her searchlight, which confused the helmsman of the *S. 126*, and the torpedo boat got under the *Undine's* bow and was struck amidship, causing the boiler of the *S. 126* to explode. The suddenness of the catastrophe explains the great number of victims. It is believed that all the missing members of the crew were killed or drowned. Several wounded officers were rescued. A number of tugs and small cruisers with divers went to the scene of the disaster. Emperor William, who had arrived at noon for the swearing in of the naval recruits, was immediately informed of the accident and after leaving the train proceeded to the imperial waiting room of the railway station, where he ordered a detailed report to be made to him. The accident occurred in the midst of a driving snowstorm and when the night was very dark. The torpedo boat sank in four minutes.

JAPAN.

List of War Vessels building or about to be laid down in the dockyards of Japan: At Kure, the battle ship *Aki* (18,000 tons), about to be laid down. At Yokosuka, the battleship *Satsuma* (18,000 tons), about to be laid down. At Kure, the armored cruiser *Tsukuba* (14,000 tons), laid down in February, 1905, and expected to be launched in February, 1906. At Kure, the armored cruiser *Ikoma* (14,000 tons), laid down in December, 1904, and expected to be launched in December, 1905. At Yokosuka, the armored cruiser *Kurama* (14,000 tons), laid down but dates are not given. At

Kure, the armored cruiser *Ibuki* (14,000 tons), about to be laid down. The "Jiji Shimpo," referring to this subject, says that the Japanese Admiralty has already elaborated a scheme of naval expansion which awaits the approval of the Diet. Although Japan has now a fairly large and efficient navy, it is recognized that some of her best ships are beginning to be out of date, and everything goes to indicate the necessity of building larger ships, with much heavier armaments and higher speed. In view of this change of plan, the *Kashima* and *Katori,* now building in England, would fall into the second place. The battleship of the immediate future will probably be one displacing 22,000 tons, with an armament of fourteen 12-inch guns, and a speed of 20 knots; while the armored cruiser will have a displacement of 15,000 tons, and a speed of 25 knots. The ships now under construction in Japan are an approximation to the new lines. The "Jiji" says: "England is now building the *Lord Nelson,* a battleship with a displacement of 18,000 tons, and an armament of four 12-inch guns; but English naval architects contemplate something much more formidable. Russia's new navy will certainly be on modern models, and Japan must live up to the time." We are informed that the new submarines, five in number, when performing evolutions before the Emperor at the naval review, proved a great success, and the authorities are about to increase the squadron. The Japanese believe that the submarine is especially suitable for their Navy, as its management demands precisely the qualities in which the Japanese officer and sailor excel, courage which never shrinks from selfsacrifice and absolute coolness in action. The destroyer *Asakaze,* was launched at the Kawasaki yard, in Kobe, at the end of October in the presence of a large company, and Captain Ommaney, of H. M. S. *Andromeda,* cut the rope freeing the vessel. The *Asakaze* is one of the five destroyers ordered by the Admiralty from this same firm. We learn further that a new factory for large guns is nearly completed in the neighborhood of Tokio, midway between Oji and Itabashi, covering an area of 100,000 tsubo (83 acres). It is said to be very complete in

every department, and fitted with the most approved appliances. Much energy has been shown in pushing on the work. Four thousand men are said to have been employed originally, but the number is now reduced to seventeen or eighteen hundred. The cost is put at two million yen, and the shops are now about ready to begin operations. It is quite evident that the Japanese are determined to maintain all the advantages they have gained. It is, however, to be regretted that political conditions should have forced them into the course of militarism instead of the peaceful development of the resources of their country.—"Engineering."

Tsukuba.—The armored cruiser *Tsukuba* was successfully launched on Dec. 26, 1905, at Kure in presence of the Crown Prince. The occasion was one of much public rejoicing, this being the first armored cruiser built entirely in Japan without aid from abroad.

The Japanese Navy After the War. — The task that confronted the little Japanese Navy at the outbreak of the recent war was simply stupendous. By the book, and on paper, it was simply impossible of accomplishment. Theoretically, by all the laws of naval strategy, that Navy should have been at least three times as large as it was to accomplish with certainty, the work that confronted it. At the outbreak of the war the naval forces of Japan and Russia in eastern waters were approximately equal; but for Japan successfully to accomplish what she set out to do, what it was absolutely necessary she should do if she were to win on sea and land, required a Navy, and this without any reference to the fleet in reserve in the Baltic, practically double the size which she had at command. To blockade a superior fleet of battleships in Port Arthur and a squadron of armored cruisers at Vladivostock; to provide transport for nearly three-quarters of a million of men from Japan to Manchuria, convoy that transport, and maintain its lines of communication so secure that the fresh reserves, the wounded, the hundreds of thousands of tons of provisions, ammunition, guns, and general army supplies, might pass to and fro without fear of interruption—all this was a task for

which a Navy double the size of that of Japan would have been considered by the naval strategist none too strong. Furthermore, the seemingly insuperable task (we are now weighing the question as it was weighed before the events of the war had opened our eyes) confronting the Japanese Navy was rendered doubly discouraging by the fact that the waste of war in ships and general war material would be, for the Japanese, irreparable, whereas the enemy possessed in the Baltic reserves a fleet that was approximately equal in power, and in its principal units more modern in type, than the one with which Japan had immediately to deal in eastern waters. So that it was necessary, not merely to defeat and destroy an enemy who by virtue of his strategical position was stronger than themselves, but the victory must be accomplished with the minimum of loss of ships—that is, if the Japanese remnant was not to be overwhelmed when the Baltic fleet reached the scene of conflict.

It is easy to be wise after the event; but at the opening of the war of 1904, the naval strategists would have told us that if Japan won out "by the skin of her teeth," and with but a pitiful remnant left of her own Navy, it would be a most brilliant feat of arms. We doubt if even among the Japanese themselves, well informed as they were as to the actual efficiency of the enemy, it was expected that the successful termination of the war would leave their Navy anything but sadly wrecked.

All the more splendid, then, are the results as they are recorded in the tabular statement which is herewith presented to our readers; "for not only did the Japanese Navy cheerfully accept and patiently bear the double burden imposed upon it, but it has emerged from the struggle actually 50 per cent. stonger than it was at the outset.

Of late, this Oriental race has shown to the western world some new and better ways of doing certain things in which the western world supposed itself to be pre-eminent. Japan has proved that it is possible for the personnel of a Navy to be so perfect in skill, discipline, and dauntless courage, that it can

not only win out decisively against an enemy numerically superior, but that it may emerge from the conflict more powerful in ships and material than it was when the opening gun was fired. The Japanese Navy performed many brilliant feats during the progress of the war; but not one of them was, in its way, more remarkable than the skill with which they recovered a whole fleet of Russian warships from the mud at Port Arthur and Chemulpo, took it over sea to Japan, and pushed forward the repairs so successfully as to make it possible for every battleship and cruiser before many months to go into commission under the flag of the Rising Sun.

Interest in the conflict, at least in the United States, died out so quickly and absolutely with the signing of the Treaty of Portsmouth, that the American people have failed to realize the profound significance of those occasional telegrams from Tokio which have appeared during the past few months, stating that this battleship or that cruiser had been refloated and taken to Japan. As a matter of fact, every such announcement meant that the Japanese Navy was receiving an addition of strength and taking a higher stand among the navies of the world, which, under ordinary circumstances, would have required four or five years for its accomplishment. We are informed by Japanese Naval officials that the damage done to the sunken fleet by Japanese mortar fire, and by explosives applied by the Russians themselves, is surprisingly small in comparison with what would naturally have been expected. Every one of the eleven battleships and cruisers that has been captured or refloated is capable of thorough repair. Although many of them are badly knocked about between wind and water, the amount of damage below the waterline is unexpectedly small, and not one of the heavy blows struck by the mine or the torpedo, or the large high-explosive shell, has impaired the integrity of the structure of the ship as a whole. Protective decks and waterline belts have done their work most effectually. Instances of penetration of the vitals of the ship are few; and the shells that did enter have wrought no damage that cannot readily be made good.

Without disparaging the skillful work done by the Japanese wrecking crews, it may be said that the salvage of the Port Arthur fleet is a splendid tribute to the genius of the naval architect. It is a complete verification of those theories of watertight subdivision and the combination of belt and deck armor, which have produced the many-compartmented modern warship. To be convinced of this, recall for a moment the prodigiously rough treatment to which this Port Arthur fleet has been exposed in the last two years. Torpedoed in the first night attack; pierced at the waterline in the offshore engagement next day; patched up temporarily by the use of wooden cofferdams; struck repeatedly by mines (in the case of one ship twice in the same spot); again repaired under emergency conditions; taken to sea and put through a fleet engagement of seven hours' duration; brought back to Port Arthur, to be finally sunk under a four days' bombardment by 11-inch shells; wrecked by heavy charges of guncotton applied within and without by the Russian officers themselves—these ships, after spending six to eight months at the bottom of Port Arthur harbor, are floated, and some of them taken under their own steam across the stormy Sea of Japan to be repaired and put in first-class shape at Japanese dockyards.

That the ships were raised at all was due largely to the elaborate system of watertight compartments. We do not know the total number of compartments in such a ship as the *Pobieda,* but how complete is the modern system of subdivision may be judged from our own battleship *Connecticut,* now building at the Brooklyn Navy Yard, which has seventy-one separate watertight compartments in the double bottom, 155 between the double bottom and the protective deck, and 101 above the protective deck, making a total of 327 in the whole ship.

In salving these vessels the Japanese divers first went down and closed all holes in the shell of the ship with timber, canvas and cement, or other material; shut all the watertight doors that were accessible; and then put powerful pumps at work to empty the water from the hull. In the case, for instance, of the battleship *Peresviet,* two salvage steamers were placed one on

each side, and their powerful pumps were assisted by a centrifugal pump and three pulsometers which were on board the ship itself. When all these pumps were working together, the enormous amount of 13,000 tons of water per hour was pumped out of the ship. This is more than the total displacement of the ship at her normal draught. The *Peresviet* was immersed 7 feet in the mud, and, as she lifted, the divers, of whom there were twelve squads, went carefully around the ship, closing all openings into the hull. From June 30 to July 2, the ship lifted 3½ feet; on July 3 she lifted 3 feet more, at which time she was immersed only about 3 feet below her normal draught. In spite of the fact that twelve holes made by 11-inch mortar shells were visible in the upper deck, of which four pierced the protective deck, and although four large holes were reported by divers as existing below the armor deck, one of which was 18 inches deep by 7 feet long, it is evident that no vital injury was done to the ship; for after being raised she was taken to Sasebo, Japan, under her own steam.

The same feat was performed with the armored cruiser *Bayan*, and the battleship *Poltava* was also found to be navigable under her own engines.

As a matter of fact, the condition of these ships, as revealed by the Japanese, confirms the statement made to the editor of this paper by several Russian officers, that the effect upon a warship of high explosives, whether exerted by the shell, the torpedo or the submarine mine, is strictly local. The plating, framing or what-not, that comes within the immediate radius of the explosive gases, is blown in or torn asunder by the energy of the gases or the flying fragments; but the structure of the ship as a whole is not affected. Neither the vessel itself nor its engines are thrown out of line, or wrenched, or twisted; and hence, the best answer to make to the increasing range of gun and torpedo, and to the hidden menace of the submarine and the mine, is to multiply the subdivision and still further localize the effect of the blow.

JAPANESE NAVY AFTER THE WAR.

Battleships—11, of 152,706 tons.

Name.	Date.	Displacement.	Original Trial Speed.	Remarks.
Kasuga	1906	16,400	18.5	Nearly completed.
Katori	1906	15,950	18.5	Nearly completed.
Mikasa	1900	15,200	18.6	
Asahi	1899	15,200	18 3	
Shikishima	1898	14,850	18.8	
Iwami	1904	13,566	18.0	Ex-Russian *Orel*.
Hizen	1900	12,700	18.8	Ex-Russian *Retvizan*.
Fugi	1906	12,500	18.5	
Suwo	1900	12,670	18.5	Ex-Russian *Pobieda*.
Sagami	1898	12,670	19.1	Ex-Russian *Peresviet*.
Tango	1895	11,000	16.5	Ex-Russian *Poltava*.
Coast-defense Ships—4, of 25,974 tons.				
Iki	1892	9,700	14.8	Ex-Russian *Nikolai*.
Okinoshima	1898	4,126	15	Ex-Russian *Apraksin*.
Mishima	1895	4,648	16	Ex-Russian *Seniavin*.
Chin Yen	1882	7,350	14.5	
Armored Cruisers—9 of 81,686 tons.				
Kasuga	1904	7,700	20	
Nisshin	1904	7,700	20	
Aso	1902	7,800	21	Ex-Russian *Bayan*.
Asama	1898	9,750	22	
Tokiwa	1898	9,750	22.7	
Izumo	1899	9,800	22	
Iwate	1900	9,800	21.8	
Adzuma	1900	9,436	21	
Yakumo	1899	9,850	21.7	
Protected Cruisers—20, of 71,741 tons.				
Soya	1899	6,500	24.6	Ex-Russian *Variag*.
Tsugaru	1900	6,630	20	Ex-Russian *Pallada*.
Chitose	1898	4,760	23.8	
Kasaji	1897	5,416	22.8	
Akasi	1897	2,657	20	
Suma	1895	2,657	20	
Akitsushima	1892	3,150	19	
Idzumi	1884	2,800	18	
Chiyoda	1889	2,450	19	
Hashidate	1891	4,277	16.7	
Itsukushima	1889	4,277	16.7	
Matsashima	1890	4,277	16.7	
Naniwa	1885	3,700	18.7	
Takachiho	1885	3,700	18.7	
Niitaka	1904	3,420	20	Completed during war.
Tsushima	1904	3,420	20	Completed during war.
Otawa	1904	3,000	20	Completed during war.
Miyako	1901	1,800	20	Sunk and re-floated.
Yayeyama	1891	1,600	20	
Chihaya	1901	1,250	21	

JAPANESE NAVY BEFORE AND AFTER THE WAR.

	1904	1906	Increase.
Battleships, tons	85,250	152,706	67,456 (80 per cent.)
Coast-defense, tons	9,350	25,974	16,624 (178 per cent.)
Armored cruisers, tons	73,886	81,686	7,800 (10 per cent.)
Protected cruisers, tons	55,301	71,741	16,440 (30 per cent.)
Totals	223,787	332,107	108,320 (48 per cent.)

In conclusion, we draw attention to the enormous gain in strength of the Japanese Navy due to the inclusion of these captured and refloated Russian ships. The work of new construction was carried on during the war to an extent that more than offsets the Japanese losses. The battleships *Kasuga* and *Katori*, the most powerful afloat, each carrying four 12's and four 10's and twelve 6's, will be completed early in 1906, or say two years from the beginning of the Russian war; and the Japanese themselves completed three 20-knot cruisers of about 3,000 tons displacement. Five battleships, three coast-defense ships, one armored cruiser and two large protected cruisers of the late Russian fleet have been added to the Navy; and it is probable that, in the case of the battleships, the Russian armament will be replaced by much more powerful guns of the wire-wound type of Japanese manufacture. The speed of these ships is high, and they may be considered, when re-armed, as being well up to date. The increase of the Japanese Navy by accretions, during and after the war is as follows: In battleships there is an increase of 67,456 tons, or 80 per cent. in coast-defense vessels, an increase of 16,624 tons, or 178 per cent.; in armored cruisers there is an increase of 7,800 tons, equal to 10 per cent.; and in protected cruisers there is an increase of 16,440 tons, or 30 per cent. The total increase of the whole Navy in tonnage alone is 108,320 tons, which is an increase of 48 per cent. on the total tonnage with which Japan went into the war. This places her ahead of Italy, and brings her into fifth place, or next in rank to the United States.—"Scientific American."

RUSSIA.

Reconstructing the Russian Navy.—The naval section of the Imperial Russian Technical Society has completed an

investigation as to the possibilities of constructing a new Russian fleet in Russian shipbuilding yards, and has arrived at the conclusion that this can be done. In five years the Baltic Works can build eight first-class cruisers, and the Galerny Island and New Admiralty Yards twelve first-class battleships. Kronstadt, with its Peter Dock, is in a position to construct armored cruisers of the improved *Bayan* type and sea-going gunboats or torpedo transports. Torpedo cruisers and submersibles may be ordered from the Neva Works and from Lange's Works at Riga. Sea-going torpedo boats (including destroyers) and coast-defense torpedo boats can be successfully built in the shipbuilding yards of Finland, Riga, Reval and Libau, which can probably also take in hand the construction of part of the torpedo transports or storeships required. The Putiley Works can complete in the same period four cruisers of the *Bayan* type, and a number of torpedo boats and gunboats. As to works in the south of Russia, it is estimated that Nicolaieff and Sevastopol can complete four battleships and cruisers of the *Bayan* type, and some torpedo boats. In regard to machinery, orders may be placed with the Baltic, the Franco-Russian, the Sormovo, the Briansk and the St. Petersburg Metal Works, the Nicolaieff Works and those of Nobel and Lessner (St. Petersburg). As to guns and torpedo gear, although precise data are not forthcoming as to the capacity of the Government works for their production, the Society is of opinion that their output might be increased to the desired extent. Private works can supply a large quantity of torpedoes and projectiles. The manufacture of armor plating presents the greatest difficulty, and in all probability some orders will have to be placed abroad. The first thing, however, which obviously has to be done is to restore internal tranquillity in Russia.—"Engineering."

Russian Torpedo Boats.—The construction of torpedo boats has been followed with the greatest interest by the Russian Minister of Marine. Russia has now sixty-two torpedo boats in course of construction, of which forty-four have been ordered by the Minister of Marine and eighteen by the Rus-

sian Maritime League. These last are to have a displacement of 500 tons each, and they are to steam at the rate of 25 knots. Of the forty-four torpedo boats which are being constructed by order of the Minister of Marine, twelve are being built in France by the Forges et Chantiers de la Méditerranée at Havre and La Seyne; the remaining thirty-two are in hand in Russia. The expression "torpedo boats" appears to be in use rather vaguely, as the forty-four ordered by the Russian Minister of Marine comprises two transports of 570 tons each. The forty-four also include two torpedo boats of the *Sokol* type, of 207 tons each, 29 of 350 tons each, and 10 of 35 tons each. The torpedo boats with a displacement of 350 tons each will be 216 feet 8 inches long by 21 feet 8 inches beam, and they are to steam at the rate of 26 knots. Four of these torpedo boats are to be delivered in the course of this year, and four more in the spring of 1906.

MERCHANT SHIPS.

Empress of Britain. — The Fairfield Shipbuilding and Engineering Company, Limited, launched on Saturday, November 11, 1905, the twin-screw steamer *Empress of Britain*, the first of two vessels being built at Fairfield to the order of the Canadian Pacific Railway Company, for their mail and passenger service between this country and Canada. The sister-ship, *Empress of Ireland*, is well under way and will be launched in January. The *Empress of Britain* is 569 feet in length over all, 65 feet 6 inches in breadth, 40 feet in depth to upper deck, 14,500 tons gross, and about 20,000 tons displacement. There are nine watertight bulkheads, all extending to the level of the upper deck. Accommodation has been provided for 310 first-class, 470 second-class, and 500 third-class passengers, all on and above the main deck, and for 270 steerage passengers on the lower deck forward. There are in all eight decks. On the boat deck, at the fore end, there is a spacious deck house for the accommodation of the captain and officers, with chart house and wheel house. The lower and upper promenade decks amidships are entirely devoted to the accommodation of first-saloon passengers. On the upper promenade deck amidships is the first saloon entrance, having a broad, handsome staircase leading to the lower promenade deck, and from there to the shelter deck. On each side of the lower promenade and upper promenade decks amidships there is a clear promenade of 18 feet 6 inches wide for the first-class passengers, with recesses in the deck house forming sheltered corners for passengers using deck chairs. The lower promenade deck extends right aft to the stern of the ship, and forms a splendid open promenade for the use of first and second-class passengers. The first-class dining saloon measures 58 feet long by 62 feet wide, and can dine the whole

complement of saloon passengers at one sitting. The second-class apartments are on the shelter deck—viz: dining saloon, social hall and smoke room, with two large entrances, with stairways leading to the upper and main decks. The dining saloon is capable of seating about 300 persons. The third-class apartments are on the shelter deck—viz: dining saloon, commodating from two to six persons, and there is a large dining saloon having revolving chairs, &c., capable of dining over 300 persons. From stairways in each compartment the passengers can reach the promenade on the upper deck, which is covered, or to the open promenade on the shelter deck. The propelling machinery consists of two sets of quadruple-expansion engines, each set having four cylinders working on four cranks, balanced on the Yarrow-Schlick-Tweedy system. The boilers for generating steam are nine in number, in two watertight compartments, with a large coal bunker between. There are six double-ended and three single-ended boilers of the ordinary multitubular type, arranged to work with Howden's forced draft. In working out the design of the vessel the utmost consideration has been given to the experience of the latest vessels that have been built for the Atlantic trade, in order that the new vessels of the Canadian Pacific Railway Company shall embody everything that can be conceived for the safety and comfort of the traveling public. The design is by Dr. Elgar, the Fairfield Company's naval architect, and the building of the vessels has been under the direction of Mr. Arthur Piers, the manager of the Canadian Pacific Railway Company's steamship lines, assisted by Captain Mowatt, the marine superintendent at Liverpool, and Messrs. J. Gillies and T. H. Baird, as resident overseers at Fairfield.—"Engineering."

Empress of Ireland.— The *Empress of Ireland,* the second of the two large steamers which the Fairfield Shipbuilding and Engineering Company, Limited, Govan, are building for the Canadian Pacific Railway Company, was launched on January 27th. The first vessel, the *Empress of Britain,* was launched in November of last year, and is now far advanced in fitting

out. She is expected to be ready for delivery in April. The two vessels are sister ships. Each is 550 feet in length, 65 feet in breadth, 40 feet in depth to upper deck, of 14,500 tons gross, 20,000 tons displacement, and 18,000 indicated horse-power. They are built to Lloyd's three-deck and shelter-deck, 100 A1 class, with forecastle and long bridge. A cellular double bottom is fitted the full length of each ship, and the vessels are subdivided into separate compartments by water-tight bulkheads, which are arranged and constructed so as to obtain every security. There are in each nine watertight bulk-heads, all extending to the level of the upper deck, and these are built in accordance with the recommendations of the Bulk-head Committee, so that with any two compartments filled with water the ship would still remain afloat. Accommoda-tion is provided on each vessel for 310 first-class, 470 second-class and 500 third-class passengers, and for 270 steerage pas-sengers on the lower deck forward. The propelling machin-ery consists of two sets of quadruple-expansion engines, each set having four cylinders working on four cranks, balanced on the Yarrow, Schlick and Tweedy system. The high-pres-sure and the first intermediate-pressure cylinders are each fit-ted with a piston valve, and the second-intermediate and low-pressure cylinders have ordinary flat-side valves, all being worked by the usual link-motion valve gear. The crank shafts are in four sections, each section being built, and together with the thrust, tunnel and propeller shafts are of forged mild steel. The propellers have each four blades of manganese-bronze, the bosses being of cast steel. The boilers are nine in number, and are in two watertight compartments, with a large coal bunker between. There are six double-ended and three single-ended boilers of the ordinary multitubular type, ar-ranged to work with Howden's forced draft, constructed en-tirely of steel, and adapted for a working pressure of 220 pounds. The double-ended boilers have each eight furnaces, and the single-ended four furnaces, making a total of sixty furnaces. The products of combustion are led into two fun-nels.—"The Practical Engineer."

Alterations on the Virginian. — The Allan Line turbine steamer *Virginian,* is about to be taken in hand by the builders, Messrs. Workman Clark & Co., Belfast, for a change of propellers, three-bladed propellers taking the place of the four-bladed ones now on the shafts. To compensate for the loss of a blade, the new propellers will have an increased diameter of one foot. The number of revolutions will also be reduced to about 300. The alteration will, it is believed, add materially to the success of the ship, for the edge friction will be greatly minimized by the smaller number of blades and the slower engine speed. The new propellers will be of the best metal, and great care has been exercised in selecting the pitch. H. M. Ship *Iris* has been subjected to a similar change of propellers, with the result that the high speed she had previously attained was increased by at least a knot.

New Italian Steamers. — The Navigazione Generale Italiana, which is now maintaining a service of four steamers between New York and Mediterranean ports, has placed contracts for four new and much larger ships, to be constructed in eighteen months at Genoa, Palermo and Ancona. These ships are intended to enter the transatlantic trade in competition with the best and fastest steamers, and it is the intention of the company to reduce the time of the voyage from New York to Naples, now requiring thirteen days. The ships will be equipped with wireless telegraph service, dining rooms for children and nurses, and passenger elevators (electric) running from the lowest berth deck to the upper promenade. Though the tonnage of the ships is not stated, it is understood here that they are to be something over 12,000 tons.

The Manxman. — A new turbine-driven steamer, called the *Manxman,* has been recently built at Barrow-in-Furness by Messrs. Vickers Sons & Maxim, Ltd., for the Midland Railway Co.'s Isle-of-Man service. This steamer comprises three sets of expansion turbines, the center shaft being driven by the high-pressure turbine, and each of the two side shafts by a low-pressure turbine, the astern driving turbines being mounted also on the side shafts. These latter take steam direct from the boiler.

The *Manxman* has a length on the waterline of 330 feet, a breadth molded of 43 feet, and a depth of 29 feet 6 inches. She has four decks, one of these—the shade deck—being entirely devoted to a promenade, while the promenade deck has a considerable width on each side for the same purpose. The dining room occupies the full width of the ship, and has seating accommodation for 100 passengers. The saloon is situated at the forward end of the promenade deck, and occupies, with the smoking room, a deck house which extends for over 130 feet of the length of the ship. On the deck below there are arranged the principal sleeping rooms in the ship. Many of these cabins have been made portable.

The heating and ventilating are by the thermo-tank system of the Thermo-tank Ventilating Co., of Glasgow. This system aims specially at insuring to all the living quarters of the ship a continuous supply of fresh air, which is not only warmed to the requisite degree, but is also humidified, so that none of the bad effects of over-drying can be felt. In cold weather the warmed air is discharged through a regulator into each apartment near the level of the ceiling; as it cools, it gradually sinks to a lower level, carrying with it any carbonic acid gas to the passageways, where means are provided for allowing it to pass outside. The circular thermo-tanks for circulating the warm air are placed on deck. The thermo-tank consists of an electric motor operating a fan which discharges air to the outside of a tube heater. The air then passes through the tubes, and comes in close contact with the heater surface, flowing thence to the main distributing trunks. Tests have shown that where the steam-heated system took three hours to attain a given temperature the thermo-tank only required fifteen minutes.

The turbine machinery was constructed by the Parsons Marine Turbine Co., Wallsend-on-Tyne. There are two double-ended boilers and one single-ended boiler, and the principal dimensions of these are, for the double-ended boilers: length, 22 feet; diameter, 19 feet 7 inches; number of furnaces, 6; diameter, 3 feet 11 inches; length of grate, 6 feet 6 inches; heating surface, 4,984 square feet; grate area, 161

square feet. For the single-ended boiler: length, 11 feet 6 inches; diameter, 19 feet 7 inches; number of furnaces, 3; diameter, 3 feet 11 inches; length of grate, 6 feet 6 inches; heating surface, 2,493 square feet; grate area, 80 square feet. The total heating surface is 12,461 square feet, and the total grate area 402 square feet. The two double-ended boilers are placed side by side in the after part of the boiler room, while the single-ended boiler is fitted in a recess in the center of the boiler room forward, with large coal bunkers on each side. The boilers are worked under forced draft, the stokeholds being closed, and four fans are located on the upper deck, and are driven by high-speed engines, supplied by Messrs. Gaul & Co., Dumbarton.

In respect of speed, the *Manxman,* is three-quarters of a knot faster than the *Londonderry,* another Midland vessel with smaller turbines taking steam at 190 pounds pressure. Two trials were made of the measured mile and the results were the following: Mean speed of two runs, 22.141 knots; boiler pressure per square inch, 192 pounds; steam in high-pressure turbine, 180 pounds; in low-pressure turbine, port, 20 pounds; in low-pressure turbine, starboard, 20 pounds; vacuum in condenser, port, 28.29 inches; vacuum in condenser, starboard, 28.4 inches; revolutions per minute, high-pressure turbine, 553; low-pressure turbine, 609; temperature of feed water leaving heater, 180 degrees Fahrenheit; air pressure in stokehold, 1.9 inches. The results for the official six-hours trial were as follows: Mean speed, 22.60 knots; revolutions, high-pressure turbine, 520; low-pressure turbine, 590; vacuum, port, 28.6 inches; vacuum, starboard, 28.4 inches. The vacuum was frequently as high as 29 inches. In this respect a great improvement has been effected by the use of a "vacuum augmenter." In it the air pumps are placed about 3 feet below the bottom of the condenser. From any convenient part of the condenser, preferably near the bottom, a pipe is led to an auxiliary condenser, generally about one-twentieth the cooling surface of the main condenser, and in a contracted portion of this pipe a small steam jet is placed, which acts in the

same way as a steam exhauster or the jet in the funnel of a locomotive, and sucks nearly all the residual air and vapor from the condenser, and delivers it to the air pumps. A water seal is provided to prevent the air and vapor returning to the condenser. The small quantity of steam from this steam jet, which is only about 1½ per cent. of that used by the turbine at full load, together with the steam extracted, is cooled and condensed by the auxiliary condenser, which is generally supplied with water in parallel with the main condenser. Condensation in a condenser takes place much more rapidly and effectually if the air is thoroughly extracted than if there is much air present, as the air seems to form a blanket round the tubes, and prevents the steam from getting free access to them. —"The Marine Review."

The Turbines of the "Carmania." — The triple-screw turbine-driven steamship *Carmania,* of the Cunard Line, which has recently completed its maiden voyage to New York, is a sister ship to the *Caronia,* and identical with her in everything outside of the engine room.

The *Carmania,* like the *Caronia,* was built on the Clyde by Messrs. John Brown & Co., a firm which as long ago as 1867 built the *Russia,* the first of the screw-propelled Cunarders to sail to the port of New York. The same firm has in hand one of the two 25-knot, 43,000-ton liners which are being built by the same company and will be placed in service early in 1907. The dimensions of the *Carmania* are as follows: Length on deck, 672 feet 6 inches; beam, 72 feet 6 inches; molded depth, 52 feet; depth from keel to roof of navigating bridge, 90 feet; depth from keel to top of funnel, 144 feet; depth from keel to top of masts, 205 feet. With full load the great ship draws 33 feet 3¼ inches and displaces 30,918 tons. She accommodates 300 first-class, 326 second-class, 1,000 third-class and 1,000 steerage passengers. Add to this a complement of officers, engineers, and crew of 710, and we get a total of 3,336 souls that can be housed, fed and comfortably transported. This is the population of many a thriving and well-equipped city in

the United States that calls itself populous. In spite of the great displacement of these ships, they have a coefficient of fineness of less than 0.7. In the 25-knot ships the coefficient will be not very much over 0.5.

The sister ship *Caronia* was commenced and completed within the remarkably short time of nineteen and one-half months; the *Carmania,* because of the novel conditions introduced by the installation of the turbine, took slightly longer. She was begun February 29, 1904, and launched February 21, 1905. In the construction of the *Carmania* 12,000 tons of steel and 1,800,000 rivets were required. Many of the plates are 1⅛ inch thick, 32 feet long, and 5½ feet in width. The keel plate is 5 feet deep and over 1 inch in thickness, and in the framing each fifth frame, in the machinery and boiler compartment, and each sixth frame forward and aft, is of web section, built up of ½-inch plates 30 inches in depth. The shell plating, for the most part, is 1 inch in thickness. The passenger accommodation is identical with that of the *Caronia,* which has already been fully described in this journal.

Interest in the *Carmania* centers, of course, in her engine room; for although the Allan Line steamships *Virginian* and *Victorian* have been running now for many months with turbine engines, the latter are small compared with those of the new ships, being of about 10,000 horsepower, whereas those of the *Carmania* on the trial trip developed nearly 25,000 horsepower. The great size of the turbines, and the necessity of familiarizing the staff with their construction and operation, led the builders to make a set of three screw marine turbines of 8,000 horsepower. A shop was set apart as an experimental station, in which the turbines were erected together with a full equipment of condensers and auxiliary machinery. The turbines were coupled to dynamos whose efficiency had been accurately determined, and thus it was possible, during the many months that the plant was under test, to gather a large amount of most valuable turbine data. Experiments were also carried out to determine the best form of stiffening for the big turbine casings, to prevent distortion

due to high-temperature steam. The result of the careful experimental investigation was shown in the very successful maiden trip to this port, on which the *Carmania* had to contend with extremely heavy gales for practically the whole voyage. As compared with the reciprocating engines of the *Caronia,* it is found that there is a saving of weight in the turbines of about five per cent. This is very much less than has been popularly supposed, although engineers have well understood that the saving would not in the larger turbines reach a very high figure. In the *Caronia* the boiler pressure is 210 pounds, in the *Carmania* it is 195 pounds per square inch. Steam pressure at the engines is 200 pounds in the reciprocating engine, and 150 pounds in the turbines. To insure the full advantages in economy of the turbine, it is necessary to have a very high vacuum; and hence the 27,030 square feet of condensing surface in the *Caronia* is increased to 32,436 square feet in the *Carmania,* while the capacity of the centrifugal pumps is increased 100 per cent.

The turbine machinery consists of three turbines, one to each propeller shaft. The high-pressure turbine is in the center, with a low-pressure turbine on each side of it. Adjoining each low-pressure turbine is a surface condenser. The high-pressure turbine is about 8 feet in diameter, and the low-pressure turbines are about 14 feet in diameter. The turbine blades, of which there are 2,115,000, vary in length from 2 to 10 inches. At the forward end of the low-pressure turbines, on the same shaft and within the same casing, are two turbines, or turbine rotors, for driving the ship astern. The total length of the combined ahead and astern turbines and their common casting is 36 feet.

From this description it will be seen that the turbine machinery of the *Carmania* is built on an imposing scale. In fact, the first impression, as one enters the engine room, is that he is facing three steam boilers, covered with Russian iron casing. As a matter of fact, the turbine equipment, with its necessarily large number of pumps and auxiliaries, occupies not much less space than the reciprocating engines of the

Caronia. The weight of each of the low-pressure turbines in the *Carmania* is 340 tons. The casings of the turbines are split longitudinally, and in order to lift the upper half for inspection, it has been necessary to install special lifting gear, which consists of a pair of massive crossheads, one at each end of the casings, to each of which is attached a heavy vertical screw, and wormwheels and gear driven by an 18-horsepower motor. The total weight of the upper half of the low-pressure turbine casing is 98 tons, and of the high-pressure casing 45 tons. The starting platform is located at the forward end of the engine room, at about the level of the top of the turbine casings. The main throttle valve, which serves all three turbines, is shown in our illustration in the front of the platform, and in the center are shown two of the three sets of controlling levers, the center one being for the high-pressure turbine, and the two outer sets for the low-pressure turbines. High-pressure steam is admitted to the forward end of the high-pressure turbine and exhausts at the after end, from which it is led back into the forward end of the low-pressure turbine, and after passing through the blades, exhausts at their after end into the two condensers. The two go-astern turbines at the after end of the low pressures run normally *in vacuo,* and do no work. When running astern they are fed live steam direct from the main throttle.

Each of the three propellers is three-bladed, the blades being of the coarse pitch and large area which have been found best adapted for turbine service. They are only 14 feet in diameter, or say about 63 per cent. of the diameter of the propellers of the *Caronia.* They are driven at an average speed of 180 revolutions to the minute, which is by far the lowest propeller speed yet attempted with turbine engines. Except for their larger size, as described above, the condensers are of the ordinary cylindrical type. They are fitted with double-cylinder dry-air pumps, with a view to obtaining the fullest possible vacuum.

Governors are fitted to each of the turbines, and they are so adjusted that any increase beyond ten per cent. in the revolu-

tions of any of the turbines shuts off the steam until the revolutions fall to the normal speed. There is also an emergency governor provided, which entirely stops the turbines should any racing take place. We may mention in conclusion that should the *Carmania* be taken up by the British Navy as an auxiliary cruiser, she would carry twelve quick-firing guns of large caliber. Moreover, two sets of steering gear are provided, one for ordinary navigation, and a duplicate set placed below the waterline, in agreement with Admiralty specifications.—"Scientific American."

NOTES ON MERCHANT SHIPS.

New York City's New Municipal Ferry System.—The inauguration of New York City's new Municipal Ferry System and its installation between the Boroughs of Manhattan and Richmond on October 25th, attracted great attention and evoked the praise and admiration of all; owing to the magnitude of the enterprize, and the exceptional and superior features embodied in the new ferryboats.

When the idea of rapid transit to Staten Island was conceived and favorably determined upon, the New York Dock Department called on Prof. Alex. J. Maclean, of Millard & Maclean, Naval Architects, to prepare the necessary designs and specifications, and how well the exacting conditions imposed have been met, and all the requirements fulfilled, is shown in these splendid boats, fully equipped with all the modern conveniences and superior to all others for like service.

The city is now running this ferry successfully with the five boats: *Manhattan, Bronx, Brooklyn, Queens* and *Richmond,* named after the five Boroughs of Greater New York. They are running on a regular schedule of fifteen-minute intervals during the busy hours of the morning and evening, and on a half-hour schedule the remainder of the day and night.

The boats are exceptionally large and commodious, being 250 feet long and 66 feet in width over guards, and with a draught of 12 feet. They are the finest and fastest ferryboats in New York Harbor, or it is safe to add, anywhere else. In a trial trip on the Cheseapeake Bay the *Manhattan* made a

speed of nearly 19 knots per hour against the tides, and every one of these five boats has made equally fast time when they were tried out.

It was first intended that all the boats should be constructed at Sparrows Point, Md., but later the Maryland Steel Co. sublet the construction of the boat *Richmond* to the Burlee Dry Dock Co., at Port Richmond, S. I. These boats are identical in design and construction, with steel hulls and wooden superstructures, each having six watertight bulkheads, which make them practically unsinkable. One distinctive feature of these boats, noticeable to the traveling public, is found in the exceptional comfort to passengers. The interior of the cabins is fitted in mahogany, and all the seats are of the same material, with ample space and arm rests. The upper cabins are of ample size, being spacious, well lighted, and the plate-glass windows are so arranged as to give an unobstructed view in all directions. Cross seats are also provided. Rubber tiling is used in the covering of the upper saloons, the lower saloons being covered with linoleum. Ample toilet arrangements are provided replete with the most modern conveniences; reliable and complete equipments of life-saving appliances are also installed.

Motive Power and Equipment. — These boats are propelled by twin steel screws, driven by two inverted, direct-acting, two-cylinder, compound engines, having crank shafts bolted together, and with steam cylinders 22½ inches and 50 inches diameter by 30 inches stroke, and develop over 4,100 H.P. at 157 revolutions per minute.

Two boiler rooms are provided, one at each end of the engine room, in which are placed Babcock & Wilcox water-tube boilers, fitted with blowers for forced draft, of the closed stokehold system.

Pumping Machinery.—The design and construction of the pumping machinery and use of steam pumps on these boats, are so exceptional as to deserve special mention, and in the equipment for each boat are the following pumps, designed and built by A. S. Cameron Steam Pump Works, of New York City:

One (1) 10 by 22 by 18 vertical twin-beam air pump;

Two (2) 10 by 6 by 13 vertical marine piston boiler-feed pumps;

One (1) 12 by 8 by 16 vertical marine piston boiler-feed pump;

One (1) 6 by 8 by 8 vertical marine piston sanitary pump;

One (1) 5 by 4 by 7 horizontal piston fresh-water pump;

One (1) 6 by 10 by 12 combined air pump and jet condenser.

The vertical twin-beam air pumps are used to pump the condensed exhaust steam from the main condensers and deliver it to the feed-water tank or overboard, being of ample size, with 10-inch steam cylinder, 22-inch air cylinder, 18-inch stroke.

The steam end consists of two independent "Cameron" vertheir steam-chest plungers connected together, thus insuring pressure and temperature common to marine work, and having their steam-chest plungers connected together, thus insuring absolute certainty of harmonious action under all conditions. At the same time, should it be desired for any reason to run either steam cylinder alone, the other may be disconnected in a very few minutes and the whole pump driven by one steam cylinder. The cylinders are connected by piping and provided with suitable valves for running either side independently, and are mounted on a suitable cast-iron plate in such a manner as to allow the lower covers to be removed, which permits the removal of the steam piston and the rod from the lower end of the steam cylinder, a noteworthy feature where the head room is limited. The steam piston-rod stuffing boxes are fitted with metallic packing. The steam pistons are secured to their rods by a taper-fit bronze nut and split pin, and have removable followers, spring and bull rings, these parts being removable without disturbing the piston-rod nuts. The followers are secured by wrought-iron studs provided with composition nuts and brass split pins. The spring rings (of which there are two in each piston) have scarfed joints and are turned off to size after being cut and sprung together. The steam cylinders and chests are heavily lagged with non-conducting material encased in Russia-iron jackets.

The vacuum end is made of four iron parts—a suction chamber, a discharge chamber and two pump barrels, the latter lined with composition. Both suction chamber and discharge chamber are substantial castings, heavily ribbed and so constructed as to permit of free and easy flow of water at every point, and special attention has been given to the entire elimination of air pockets. The foot and head valves are on removable composition valve plates and the valves are the "Kinghorn" type, consisting of thin brass discs. The areas through the valve seats are unusually large, and special provision has been made to prevent any of the parts from unscrewing or becoming loose. The pump pistons are of composition and contain the bucket valves and are fitted with "Tucks" packing. The vacuum piston rods are of Tobin bronze, and are attached to the piston by taper fit and nut and split pin.

The vacuum-cylinder stuffing boxes are of ample depth and fitted with soft packing, and both the cylinder head and the gland provided with removable bronze thimbles. Ample hand holes are provided in the working barrels and chambers for inspection or repair of the valves.

The steam cylinders rest on a substantial plate supported by two heavy, rigid cast H-frames, the horizontal bars of which support the bearings for the rocker shaft and, at the same time, thoroughly stiffen the pump longitudinally. The rocker shaft is of steel, and works in large adjustable bearings. The beams are of steel, and amply strong to drive both pump pistons when one steam cylinder is disconnected. The links are of the standard marine type, consisting of steel rods and solid bronze ends working on steel pins of large bearing areas. The crosshead is a steel casting, the piston rod being secured in it by taper fit and taper keys (which are held by set screws.) This allows a quick and easy disconnection of either side of the pump from its steam cylinder. To prevent the side thrusts of the links from coming on the piston rods and wearing out the stuffing boxes, the crosshead is provided with adjustable bronze shoes which bear on a heavy guide bar, taking up all the side thrust.

To permit constant running, a continuous oiling system is provided consisting of stationary sight-feed oil cups with separate pipes leading to all working parts. Drain cocks are located at all necessary points, affording quick and easy drainage; suitable provision is made for holding up the vacuum cylinder covers, pistons and followers when packing the pump. All parts requiring to be removed or handled are provided with forcing screws and lifting bolts.

The boilers are fed by "Cameron" vertical boiler feed pumps of the marine piston type. The main feed pumps, size 12 by 8 by 16, are installed in the engine room. There are also two auxiliary feed pumps, one in each boiler room, of the same type but of the size 10 by 6 by 13.

The sanitary pumps, size 6 by 8 by 8, are of the "Cameron" vertical marine type, having entire bronze water cylinders, and are used for delivering sea water into a 100-gallon pressure tank, in which a constant water pressure of 20 pounds is maintained.

The fresh-water pump, size 5 by 4 by 7, is of the "Cameron" regular horizontal piston pattern, and supplies the drinking fountain, coolers and toilet rooms, maintaining an even water pressure of 20 pounds.

There is also installed in the engine room of each boat a 6 by 10 by 12 "Cameron" combined air pump and jet condenser.

The steam ends of all of these vertical marine pumps are of the standard vertical "Cameron" type, designed to withstand the high steam pressure and temperature common to marine work. The steam mechanism consists of four stout pieces only, none of them exposed to injury. The plunger is reversed by means of two plain tappet valves, and it is the only inside valve gear that is absolutely reliable. The steam cylinders and chests are lagged with non-conducting material, securely encased in sheet-iron jackets.

The water cylinders are fitted with removable composition bushings, which may be readily and quickly removed or re-

placed without dismounting the pumps or disturbing the piping.

The water-valve chest is cast in one piece with the cylinder, reducing the number of gaskets to the minimum, and special provision is made against air pockets. The pump valves are of the "Cameron" type, the seats being driven in on a taper, and one valve stem passing through both the suction and discharge valves, valve seats, springs and spring holders, making it impossible for any part to become loose or disarranged; yet by the removal of this stem all these parts may be taken out. The lift of the valves is limited by a positive stop. One bonnet covers the entire front of the chest, and when removed exposes its whole interior and gives access to every part of it. The area through the valves and passages is made very large to allow the pumps to run at a high speed, but the number of valves has been kept as few as possible to reduce the liability of breakage and to facilitate replacing them as quickly as possible. A large air chamber is placed directly over the valve chest where it is most efficient and available, by not taking up any floor space and not interfering with the piping.

The steam and water ends are connected rigidly by four heavy steel rods secured directly to the cylinders by means of substantial lugs, thus allowing the cylinder heads to be removed or replaced without dismantling the pump.

The steam piston is secured to the rod by a taper fit and a bronze nut and split pin, and has removable follower, spring and bull rings, these parts being removable without disturbing the piston-rod nut. The follower is secured by wrought-iron studs, provided with brass nuts and brass split pins. The spring rings (of which there are two) have scarf joints, and are turned off to size after being cut out and sprung together. The water piston and follower are of composition, secured to the rod by screwing home against a shoulder and with a jam bronze nut and split pin.

The piston may be packed from the upper end of the water cylinder. The ring follower is secured by Tobin bronze studs, nuts and split pins. The piston rod is made amply large and is

in two parts, the steam end being of steel and the water end being of Tobin bronze; they are fastened together by screwing into a clamp nut made in halves, facilitating disconnecting the rods. Either piston and its rods can be taken out between the tie rods without dismounting the pump or disturbing the piping, which is a decided advantage where the head room is limited. The steam-cylinder stuffing box is bushed with bronze thimble and is fitted with metallic packing, suitable for high steam pressure and temperature. The water-cylinder stuffing boxes are of ample depth and are fitted with soft packing, and are bushed with bronze thimbles, as are also the glands.

In all of these pumps the pipe openings are flanged, and the drain cocks are fitted to all points requiring drainage in both cylinders and chests. All parts that have to be removed are provided with forcing screws and lifting bolts, and suitable provision is made for holding up the pistons, rod and water-cylinder cover when packing the water piston. Sight-feed lubricators and complete sets of wrenches are provided for all these pumps, and every detail has apparently been worked out with great care and exactitude; and in the aggregate they represent a complete and perfect pumping equipment.

A description of the trial of the *Brooklyn* will be found on page 1212, Vol. XVII, of the JOURNAL.

E. M. CORYELL, *Associate.*

OBITUARY.

WILLIAM HENRY SHOCK.

REAR ADMIRAL G. W. BAIRD, U. S. N.

Chief Engineer William Henry Shock, U. S. N., died at his residence in Washington, D. C., on December 18th, 1905, in the 85th year of his age.

He was born in Baltimore, June 15, 1821, of American parents whose family had been identified for generations with the State of Maryland. He received the best education that the schools of Baltimore afforded, and was apprenticed with the firm of Watchman & Bratt, ship and engine builders. He served a three-years' apprenticeship, divided between the pattern shop, the machine shop and the draughting room, and entered the Navy through competitive examination as a Third Assistant Engineer, January 18th, 1845, at a time when there were but few steamers in the Navy. His first service was on board the *General Taylor,* the *Princeton* and *Spitfire* during the war with Mexico, and he participated in the capture of Tampico, Alvarado, Tuskan, Placotalpan and Vera Cruz. He served on board the *Mississippi* with Commodore Perry. He was promoted to a Second Assistant Engineer in 1847, and served on board the paddle-wheel steamer *Engineer.* In 1848 he was promoted to First Assistant Engineer, and was ordered in charge of the engineering department of the *Legare,* a Coast-Survey steamer, where he remained until 1850, when he was ordered as superintendent of the construction of the machinery of the *Susquehanna,* at Philadelphia. In 1851 he was promoted to be Chief Engineer and was ordered to Boston to superintend the building of new machinery for the *Princeton.* In 1853 and 1854 Mr. Shock was appointed as Inspecting Engineer of all ocean-going steamships carrying U. S. mails,

in addition to being Chief Engineer of the *Princeton* on the Home Squadron.

In 1855 he was ordered to West Point to superintend the construction of the famous steam frigate *Merrimac,* after which he served in that fine ship as her Chief Engineer during her cruise in European waters. He served on board the *Powhatan* from 1857 to 1860. From 1860 to 1862 he was President of the Board to examine engineers for admission to and promotion in the Navy. In 1862-3 he superintended the construction of river monitors at St. Louis, and from 1864 to 1865 he was Fleet Engineer of Admiral Farragut's fleet in the Gulf of Mexico, and acquitted himself well in those strenuous times. Subsequently he was Chief Engineer, first of the Boston Navy Yard, then the Washington Yard, and in 1869 was Fleet Engineer of the European Squadron. On his return to the United States he was made Inspector of Machinery afloat, and served as a member of the Board of Visitors at the Naval Academy. He was temporarily Chief of the Bureau of Steam Engineering in 1870, and again in 1871, for which service he received a letter of thanks from the Department. In 1873 he was sent on a tour of inspection in Europe to collect information concerning dock yards and marine-engine plants, and was a representative of the Bureau at the Vienna Exposition, and, by the President, was appointed as one of the American judges of awards. He was commissioned as Chief of the Bureau on March 3d, 1877, in which position he served until 1883, when he was retired.

This record covers a period of great activity and progress in the science of Naval Engineering. Beginning with the little side-wheel vessel *General Taylor,* he continued on board the *Princeton,* the first screw-propeller ship in the Navy, if not the first in American waters. His experience on the great side-wheelers *Mississippi, Powhatan* and *Susquehanna,* as well as in the construction of their machinery, was of immense advantage to him. In that day there was little or no literature on the subject of marine engineering; it was in a period largely when steam hammers were unknown, and when it was

considered a great feat to cast a 60-inch cylinder. The contractors, in that day, leaned on the Inspectors for advice and information.

The engines of the SIX FRIGATES (the *Merrimac* class) were the largest and most powerful screw-propeller engines ever undertaken in this country up to that time. The opposition to engines and the engineers was at that time so serious that it was an embarrassment, if not an obstruction. But Mr. Shock was always abreast of the times. During the civil war a very small propeller boat was captured by the *Calhoun* and was sent to New Orleans; its boiler was lost overboard en route. Mr. Shock, with the assistance of Second Assistant Engineer S. L. P. Ayers (an excellent draughtsman), quickly designed and had built a proper boiler for the boat, which became the pioneer steam launch of the Navy. He then built machinery for a cutter and a launch, which had been saved from the wreck of the *Mississippi;* the former was used as a steam cutter for the *Pensacola,* and the latter (the *Loyal*) was used as Admiral Farragut's barge.

The ships of the Navy had been proverbial for their cleanliness—their very decks were so admired for their whiteness. To see the time-honored quarter deck soiled by the black, soiled shoes of the coal heavers was as serious an objection as to see the main topsail begrimed by soot. Whether the engine had come to stay was a burning question. With it came the elimination of the romance of the seaman's life; with it came hard, practical sense, hard to combat and hard to swallow. Even after the termination of the civil war the text book on seamanship used at the Naval Academy queried whether or not "coal whips would outlive tacks and sheets."

Mr. Shock, in his Christian meekness, pursued his straight path (narrow though it be) and lived to design the excellent up-to-date compound engines, and he lived to see mastless ships which were successfully navigated without dependence upon tacks and sheets.

In his day and generation every engineer was self made. There was no literature on the subject when he began. Every

engineer made his own text books from his own experience. Mr. Shock's admirable book on Steam Boilers, published about 1882, is still standard.

During all his life he was a consistent Christian, a devout member of the Methodist Episcopal Church. He was an indulgent husband and a kind father. He is survived by but one son, a number of grandchildren and great grandchildren. He was a member of the American Society of Naval Engineers, of the Society of Mechanical Engineers, and the Society of Freemasons.

CAPTAIN OF ENGINEERS, JOHN WILKIE COLLINS, U. S. R. C. S.

Captain John Wilkie Collins, late Engineer-in-Chief of the Revenue Cutter Service, died in Amityville, Long Island, N. Y., February 2, 1906. Captain Collins was born in New York City January 29, 1845. After attending the public schools of his native city and serving an apprenticeship in the shops of the South Brooklyn Steam Engine & Boiler Works he was, at the early age of nineteen years, appointed as a Third Assistant Engineer in the United States Navy and at once assigned to duty on the United States frigate *Wabash*. While attached to that vessel he saw considerable active duty and took part in both battles of Fort Fisher. He was subsequently transferred to the gunboat *Tacony*, which vessel was engaged in duty in the sounds of North Carolina until the close of the war. Receiving an honorable discharge from the volunteer navy he was commissioned a Second Assistant Engineer in the Revenue Cutter Service on April 7, 1866. He rapidly rose through the various grades in this service until he was made a Chief Engineer on July 18, 1878. In the Revenue Cutter Service he has performed active duty on many of its vessels stationed on the North and South Atlantic coasts, and on the lakes. He was, during this period, twice assigned as Chief Engineer of the cutter *Andrew Johnson*, stationed at Milwaukee, Wis., and is therefore well known to many people along the Great Lakes. Up to January, 1892, the position of

Consulting Engineer of the service has been filled by an engineer in civil life. Chief Engineer Collins was prior to that time engaged in duties as an engineer member of the International Marine Conference. So well were these duties performed that Secretary Foster selected him from the entire list of Chief Engineers of the service and detailed him as Consulting Engineer, which thus made the subject of this sketch the first officer of the service to hold that responsible position. This selection was subsequently ratified by the Act of Congress approved July 31, 1894, creating the position of Engineer-in-Chief of the service. The work of this office was so greatly increased on account of the building of new vessels that Congress, in 1897, created the grade of Captain of Engineers, which carried with it the pay and emoluments of a captain in the line. President McKinley appointed Captain Collins to this new position on July 26 of that year. He thus became the first marine engineer in the employ of the United States Government to have what is commonly known as positive rank, a desideratum long striven for and subsequently attained by the engineer officers of the Navy.

At the time Captain Collins was made the Consulting Engineer of the service there was no revenue cutter afloat having a triple-expansion engine, although at that time this type had been almost universally adopted in the merchant service. The first machinery constructed under his directions, that of the *Hudson,* was fitted with a triple-expansion engine and a water-tube boiler. This, it will be remembered, is the vessel which performed such gallant and meritorious service at the battle of Cardenas, when she rescued the torpedo-boat *Winslow* from almost certain destruction by the Spanish batteries.

DAVID MAXSON GREENE.

David Maxson Greene, a Member of the American Society of Naval Engineers, and one of the best known civil engineers in this country, died at his home in Adams, Jefferson County, N. Y., November 9, 1905.

Mr. Greene was born in Brunswick, Rensselaer Coun-

ty, N. Y., July 8, 1832. In 1835 his parents moved to Adams, Jefferson County, N. Y., where he received his early education in the district schools and at Adams' Seminary. In 1850 he entered the Rensselaer Polytechnic Institute, where he completed by great industry a three-year course in one year, graduating in 1851 with the degree of Civil Engineer. He was an instructor at the Institute during the following year, and was later appointed to a subordinate position in the State Engineering Corps, then engaged in enlarging the Erie Canal. In 1853 he went to Ohio and engaged on a survey of the Cleveland, Lorain and Wheeling Railroad. The following spring he went to Indiana, where he had charge of a division of the Logansport and Northern Indiana Railroad. He was obliged to return East in the fall on account of ill health. In 1855 he was appointed an instructor in railroad surveying at the Rensselaer Polytechnic Institute and was shortly afterward elected professor of geodesy and topographical drawing. In the spring of 1856 he was sent to West Point and took a special course in topographical engineering under Gen. Thomas H. Neill.

IN THE UNITED STATES NAVY.

He continued to hold the professorship in Troy until 1861, when he applied for admission in the United States Navy. Attached to the United States frigate *Susquehanna*, he participated in the attacks and capture of the forts at Port Royal and Hatteras Inlet. He was engaged in blockade duty off the Atlantic coast and in the Gulf of Mexico. In September, 1862, he was detached from the *Susquehanna* and ordered to the United States Academy at Newport, R. I., as assistant professor of natural and experimental philosophy and instructor in steam engineering. In 1865, at his request, he was detailed for duty as assistant to the Chief of the Bureau of Steam Engineering in the Navy Department at Washington, D. C. For three years he was engaged in planning and conducting elaborate experiments on the physics of steam and superintended the erection of a marine engine, then the largest in exis-

tence. Early in 1868 he was appointed a member of the Treasury commission to examine and test the devices for determining the product of the distilleries of the country. This commission came to be known as the whisky commission. In the spring of the same year he entered the service of the city of Albany as Chief Assistant Engineer in charge of surveys for the extension of the waterworks. In February, 1869, he was detailed as First Assistant Engineer in charge of the United States steamship *Narrangansett* for a cruise in the West Indies, but yellow fever broke out in May, and the ship was sent back into Northern waters and was detained in quarantine at the Isles of Shoals, near Portsmouth, N. H., for two weeks. In September he was detailed as First Assistant Engineer in charge of the *Frolic,* the Port Admiral's vessel in New York harbor. He reported for duty and at the same time tendered his resignation from the naval service.

Professor Greene returned to Troy and entered on the general practice of his profession. Early in 1870 he was employed to examine the Ottawa River Canal and report to the Canadian Parliament on the probable effect of the sawdust and mill refuse discharged by lumbermen at Ottawa. In 1871 he examined the plans for a water supply for the city of Troy and advocated the pumping system, which, after a long controversy extending over several years, was finally adopted. About this time he was elected Engineer of the New York State Commission to test devices to substitute steam for animal power on the canals. In January, 1874, he was appointed Engineer in charge of the Eastern Division of the State canals. In July of the same year he was appointed Deputy State Engineer, in which office he continued until 1877, when he returned to Troy and resumed general engineering practice. In August, 1878, he was elected Director of the Rensselaer Polytechnic Institute, a position that he held for thirteen years, resigning in 1891. During that period he was engaged in a general engineering practice extending to eleven States, the District of Columbia and Canada. In 1879 and 1880 he was Chief Engineer of the Troy waterworks extension. He had a wide ex-

perience in the courts, as an expert witness on hydraulic and steam engineering. Professor Greene was a School Commissioner of Troy from 1873 to 1876.

AUTHOR OF ENGINEERING WORKS.

Professor Greene was the author of the following: "Notes On Steam Engineering" (1886); "Method of Laying Out Easement Curves On Railroads" (1891); "The Fly-Wheel as a Regulator" (1890); "Manuscript Notes On Map Projections," "Practical Hydraulics" and railroad notes for the use of his students.

BOOKS RECEIVED.

"MECHANICAL PROCESSES."--A Practical Treatise on Workshop Appliances and Operations for the Instruction of Midshipmen at the U. S. Naval Academy.—By JOHN K. BARTON, Commander, U. S. Navy, Head of Department of Marine Engineering and Naval Construction.

While there are various publications treating of each branch separately, there are but few that take up the different operations as a whole in an engineering plant, and these are bulky and unfitted for the course of instruction in engineering. They are also of foreign authorship and are largely descriptive of foreign tools and practice. In consequence of these conditions, the author has produced in this volume a book directly suited to the methods employed in the instruction of midshipmen.

All unnecessary matter has been avoided, and the student is given, as briefly as possible, the information needed to correctly understand, and efficiently use the tools and material with which he is daily brought in contact in his practical work.

This volume will also prove particularly useful to the line officers of the service, all of whom have engineering work to do from time to time, either with the main engines, torpedoes, dynamos or guns, or on Boards of Survey to pass judgment on breakages or casualties, to judge the suitability of methods proposed for repairs and alterations, and to make the estimates for the same.

The book is arranged in thirteen chapters, with an appendix, as follows:

Chapter I.—Construction Materials and their Production—Iron.
II.—Steel and its Manufacture—Non-ferrous Metals and Alloys.
III.—Wood-Working Shop—Pattern Making.

IV.—Molding and Casting.
V.—Steel Castings—Brass Founding.
VI.—The Forge.
VII.—The Machine-Shop.
VIII.—Vice Work and Tools.
IX.—Machine Tools—The Screw-Cutting Power Lathe.
X.—Lathe Tools and Attachments.
XI.—Other Forms of Lathes and Machine Tools.
XII.—Boring and Turning Mills and Milling Machines.
XIII.—The Boiler-Shop and Equipment.
Appendix.—Data and Useful Information.

Published by the U. S. NAVAL INSTITUTE, NAVAL ACADEMY, ANNAPOLIS, MD., 8vo, 356 pages, full cloth. Price $4.00.

DIE SCHIFFSSCHRAUBE (The Screw Propeller), by ALBERT ACHENBACH, Professor in the Royal Technical School at Kiel. Price 10 Marks. Kiel, Germany. ROBERT CORDES.

This work, which the Introduction tells us is to be in three parts, and of which only Part I is now under consideration, is intended as a handbook for practical use.

It begins with the usual illustration of the generation of a helix and of helical surfaces, gives a short historical description of various propellers, drawn chiefly from Baurne, and continues through the various kinds of screws of uniform and of expanding pitch, giving the principles of their construction and drawings of them. Illustrations are also given of how to sweep up the surfaces of screw blades, how to measure the pitch and set the blades, and how to draw the complete propeller. The volume concludes with a description and drawings of the Zeise and Niki screws.

This part, comprising 82 pages of text and 25 plates, gives evidence of much care in its preparation. The general method followed is practical, and the matter is so well arranged that the work should prove a valuable addition to the literature of the subject, even to those who have but a slight knowledge of German.—R. S. G.

ASSOCIATION NOTES.

The regular annual meeting of the Society was held at the Navy Department, Washington, D. C., on December 28, 1905, for the purpose of electing officers for 1906.

A count of the votes received resulted in the election of the following officers:

President, Commander A. B. Canaga, U. S. Navy.

Secretary-Treasurer, Lieutenant Chas. K. Mallory, U. S. Navy.

Members of the Council, Commander W. M. Parks, U. S. Navy; Commander R. S. Griffin, U. S. Navy; Lieutenant Commander B. C. Bryan, U. S. Navy.

SPECIAL MEETING OF THE COUNCIL.

At a special meeting of the Council, held January 2, 1906, Lieutenant Chas. K. Mallory, U. S. Navy, tendered his resignation as Secretary-Treasurer, and Lieutenant Commander Theo. C. Fenton, U. S. Navy, was elected to fill the vacancy thus created.

The following is the statement of the Secretary-Treasurer for the year ending December 31, 1905:

Received 1905:	
Dues and subscriptions	$3,932.34
Advertisments	1,870.60
Sales, JOURNALS and reprints	443.86
Interest on deposit	131.84
Refund of express charges	.30
Total receipts, 1905	6,378.94
Balance received, January 3, 1905	6,010.12
	$12,389.06

Expended:

Printing, JOURNALS and reprints	$3,847.33
Engraving and cuts	509.24
Salary, Secretary-Treasurer	900.00
Preparing and printing Index	984.40
Prize essay and medal	109.00
Purchase of JOURNALS	113.50
Printing circulars and stationery	111.50
Postage, expressage, incidentals	67.46
Total expenditures, 1905	6,642.43
Credit balance, January 1, 1906	$5,746.63

This statement was audited by a committee of the Council and found to be correct.

THEODORE C. FENTON,
Lieutenant-Commander, U. S. Navy,
Secretary-Treasurer.

A. B. CANAGA, Commander, U. S. Navy,
President.

BY-LAWS OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS.

1. The Association shall be known as the AMERICAN SOCIETY OF NAVAL ENGINEERS.

2. The object of the Society shall be to promote a knowledge of marine engineering and naval architecture by reading, discussing and publishing papers on professional subjects; by bringing together the results of experience acquired by engineers in all parts of the world, which, though valueless when unconnected, tend much to the advancement of engineering when published together in the JOURNAL of the Society; by publishing the results of such experimental and other inquiries as may be deemed essential to the advancement of the science; and historical events in the lives of engineers.

3. The officers of the Society shall be a President, a Secretary-Treasurer, and a Council, all of whom shall be elected annually.

4. The Society shall be composed of Members, Associates and Honorary Members.

5. Officers of the Line and Construction Corps, and ex-officers of the Engineer, Line and Construction Corps of the Navy, and officers and ex-officers of the Revenue Cutter Service shall be eligible as Members.

6. Persons in civil life whose knowledge of engineering is such that they can co-operate with naval engineers in the promotion of professional knowledge, or who are intimately connected with the engineering profession, shall be eligible as Associates.

7. The Secretary of the Navy, the Assistant Secretary of the Navy, the Chief of the Bureau of Steam Engineering of the Navy Department, all ex-Chiefs of the Bureau of Steam Engineering, the prize essayist of each year, and such other persons as the Society may elect, shall be Honorary Members.

8. Members shall be admitted upon application and payment of the annual subscription.

9. An Associate may be admitted upon application in writing and payment of the annual subscription, provided his application have the recommendation of a member, and receive the approval of a majority of the council.

10. Associates shall be entitled to all the privileges of Members except voting and holding office.

11. To be eligible as Honorary Member the candidate must receive the unanimous vote of the Council before his name shall be presented to the Society; and the favorable vote of two-thirds of the Members voting shall be necessary for election.

12. The direction and management of the affairs of the Society, and the editing and publishing of the JOURNAL, shall be vested in a Council composed of five members, the President and the Secretary-Treasurer being members *ex officio*. The Council may appoint one of its number Librarian.

13. The President shall exercise the usual duties of that office.

14. The Secretary-Treasurer shall conduct the correspondence of the Society and its financial transactions. He shall submit an annual statement of the receipts and expenditures of the Society, which shall be audited by three other members of the Council.

15. The Council shall have authority to fill vacancies which may occur during the year.

16. Subscriptions and all matter intended for publication, shall be sent to the Secretary-Treasurer; but no paper shall be read before the Society at the annual meeting, nor anything published in the JOURNAL, without the approval of the Council.

17. In deciding matters pertaining to the Society, none but members shall be entitled to vote; to constitute a quorum for business, nine members must be present.

18. The Society shall meet annually in the City of Washington, and at such other times and places as meetings can be conveniently arranged.

19. The annual subscription shall be five dollars, payable in advance.

20. Copies of the JOURNAL may be sold by the Society at a price which shall be fixed by the Council.

21. Absent members voting on any proposition must state over their signatures whether they are for or against the proposition.

22. On the first Tuesday in October a meeting of the Society shall be held for the consideration of nominations to offices falling vacant under the rules. Nominations to office must be made in writing, and may be made by any member, whether present at the meeting or not. The consent of the nominee to accept office if elected must be assured. A necessary qualification of all nominees, besides full membership, is residence in or near Washington during the term of office.

23. After the nominations are all made, lists of the candidates for the different offices shall be prepared by the Secretary-Treasurer, and mailed to all members of the Society. Each voting list must be signed, and must show under the different offices the name of the person voted for. Each voter will enclose his ballot in a sealed envelope, which will be sent to the Secretary-Treasurer previous to the annual meeting. At the annual meeting the envelopes will be opened and the votes counted.

24. Nothing in the preceding section shall preclude the name of any candidate from being put in nomination for more than one office. In case of a tie, the presiding officer shall decide. Voting by proxy is prohibited.

25. In the event of a candidate receiving the greatest number of votes for more than one office, he shall decide which he will accept.

26. Any office falling vacant in the course of the year owing to the death, resignation or removal from Washington or vicinity of the incumbent, shall be filled by vote of the members of the Council remaining.

27. Motions to amend these By-laws, or to make new By-laws, must be in writing, and must lie over at least three months, during which time notice thereof shall be sent to all members; and the assent of two-thirds of the members voting shall be necessary to amend or adopt.

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Wills, Alexander.....Care N. N. S. & D. D. Co., Newport News, Va.
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Winant, Wm. E.....1912 13th street, N. W., Washington, D. C.
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Wolvin, A. B.....Zenith Transit Co., Duluth, Minn.
Wood, S. L., N. N. S. B. & D. D. Co.....232 49th street, Newport News, Va.
Woolson, H. T.....Care Gas Engine and Power Co., Morris Heights, New York, N. Y.

Yarrow, A. F., Naval Architect and Mechanical Engineer, Isle of Dogs, Poplar, London, England.

Zeiter, Fr., Marine Engineer.....Bülow street, 22, Bremen, Germany.

SUBSCRIBERS.

Admiral Ahmed Pasha, M. I. Mech. E., Engineer-in-Chief,
I. Naval Arsenal, Constantinople, Turkey.
Allen, Herbert F. L.....Colorado Building, Washington, D. C.
Ansaldo, (Gio) Armstrong & Co.....Sestri Ponente, Genoa, Italy.
Atlantic Works, The.....East Boston, Mass.

Baltimore Polytechnic Institute.....Baltimore, Md.
Bauermeister, F.....49 Gordon street, Glasgow, Manchester, Eng.
Benedetti, Vittorio De, "Il Macchinista Navale,".....via S. Giuseppe 21, Naples, Italy.
Bibliothek der Kgl. Technischen Hochschule.....Dresden, A. 14, Germany.
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Bonetti, Santo, Ingineur.....Sestu Levante, (Pila), Italy.
Boston Public Library.....Boston, Mass.
Boutakoff, Alexander G., Commander, I. R. Navy, Russian Embassy,
1325 M street, N. W., Washington, D. C.
Briner, E. Amandus.....438 Queen Lane, Germantown, Philadelphia, Pa.
Bureau Steam Engineering Library.....Navy Department, Washington, D. C.
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Cammermeyer, H. Boghandel.....41-43 Carljahans Gade, Christiana, Norway.
Carnegie Library, Periodical Department.....Schenley Park, Pittsburg, Pa.
Carnegie Library, Mechanical Engineering Dept.....State College, Pa.
Cavalcanti, Vital Brandao, Engenheiro Naval, Director das Officinas de Machinas,
Arsenal de Marinha, Pará, Brazil
Cordes, Robert, Buchhandling.....Brunswicker Strasse 52a, Kiel, Germany.

Cornell University Library.....Ithaca, N. Y.
Cramp, Wm & Sons' S and E. B. Co., Engine Department.....Philadelphia, Pa.
Crucible Steel Co. of America.....Pittsburg, Pa.
Cruickshank, Alex......Naval Proving Ground, Indian Head, Md.

Delaunay, Belleville & Cie.....St. Denis, Seine, France.
Department of Marine Engineering and Naval Construction,
U. S. Naval Academy, Annapolis, Md.
Department Steam Engineering.....(via San Francisco), U. S. Naval Station, Cavite, P. I.
Department Steam Engineering.....Navy Yard, Mare Island, California.
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Navy Yard, New York, N. Y.
Direzione delle Costruzione Navali del 2 Dipartimento Marittimo.....Naples, Italy.
Dupagnier, Mr. L.....9 Avenue de la Gare, Chalon-sur-Saone, Saone et Loire, France.
Düsseldorf, Ratinger Röhrenkesselfabrik, Vormals Dürr & Co., Düsseldorf-Hafen, Germany.

Ecole d'Application du Genie Maritime.....140 Boulevard Montparnasse, Paris, France.
Engineering College, Department of Naval Architects,
Tokyo Imperial University, Tokyo, Japan.
Engineering College, Mechanical Engineering Department,
Tokyo Imperial University, Tokyo, Japan.
Engineers' Club, Japanese Naval.....Care Kan-Sei-Shibu, Navy Department, Tokyo, Japan.
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43 Gerrard street, Soho, London, England.
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Care C. B. Richards & Co., 31 Broadway, New York, N. Y.
German Embassy.....Washington, D. C.
Gildner, A. W., Jr......213 34th street, Newport News, Va.
Great Seal Patent Office Library.....Southampton Buildings, Holborn, London, England.

Hall, Thos., Superintending Engineer, Mallory Line.....Pier 20, East River, New York, N. Y.
Hardy, J. W.....Casilla 51, Valparaiso, Chili.
Harvard University, Engineering Library.....Cambridge, Mass.
Hideo Takeda, First Class Chief Engineer,
Care Kaigun-Rentan-Seizojo, Tokuyama-Machi, Yamaguchiken, Japan.
Hughes, Thomas.....10 Angerstein Road, North End, Portsmouth, England.

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(By closed mail via England.)
Ishii, Frank N.....279 6th street, San Francisco, Cal.

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Judge, Jno. P.....General Electric Co., 227 E. German street, Baltimore, Md.

Kaigunk, Daigak-o.....Naval Staff College, Tsukiji, Tokyo, Japan.
Kamo, J., I. Engineer Captain, I. J. N.....Naval Engineering College, Yokosuka, Japan.
Kan-Sen-Kyoku (Marine Bureau), Communication Department.....Tokyo, Japan.

Karljohansverns Verft.....Horten, Norway.
Kazama, T.....10 Nishikatamachi, Komagomé, Hongoku, Tokyo, Japan.
Kölner Bezirksverein Deutscher Ingenieure.....Köln, Germany.
(Lesezimmer Bürgergesellschaft.)
Kowji, Ito.....Karasumaru Marutamachi, Kioto, Japan.
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L. Dupagnier, Ingenieur, Chantiers Scheidner et Cie.,
Chalon Sur Saone, Saone et Loire, France.
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Lynch, S. M, Purchasing Agent, National Tube Co.....Frick Building, Pittsburg, Pa.

Manning, Maxwell & Moore.....85, 87, 89 Liberty street, New York, N. Y.
Manufacturers' Advertising Bureau.....126 Liberty street, New York, N Y.
Mechanics Institute.....31 Post street, San Francisco, Cal.
Milner, J. W., Engineer Lieutenant, R. N.....Upper Cleasby Rd., Menston, Leeds, England.
Ministero della Marina, Direzione Generale, Costruzione Navila.....Roma, Italy.
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Miyakawa, K.....220 Azabu Honmura Machi, Tokyo, Japan.
Murray, G. W., Engineer Commander, R. N.....Admiralty, Whitehall, London, S. W. England.

Nakajima, Yosohachi, I. J. N.....68 Inaoka, Yokosuka, Japan.
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Riggs, F. B.....Hotel Washington, Seattle, Wash.

Saito, M., Lieutenant Constructor, I. J. N.,
Care Vickers Sons & Maxim, Barrow-in-Furness, England.
Sakai, Geo. K.....279 6th street, San Francisco, Cal.
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Senhor Don Luis da Cunlia de Marrcellos, Director Technics de Arsenal de Marinha,
Lisbon, Portugal.
Sneddon, J. P., Superintendent, Stirling Co.....Barberton, Ohio.
Spett, Sotto Direzione delle Construzioni Navali del R. Cantiere,
Castellammare di Stabia, Napoli, Italy.
State Library.....Sacramento, Cal.
Steiger, E., & Co.....Newspaper Box 298, New York, N. Y.

Taguchi, Kozo, Lieutenant.....Kaigunsho, Tokyo, Japan.
Tams, Lemoine & Crane.....52 Pine street, New York, N. Y.
Tappen, A. B......263 W. 73d street, New York, N. Y.

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Thomas, H. E., Schiffbau Inginieur.....Grolmanstrasse 21, Charlottenburg, Germany.

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(To the Director, Engine Department, Naval Construction Works.)
Viennot Advertising Agency.....524 Walnut street, Philadelphia, Pa.
Virgilio Giacomuzzi.....Sestri-Ponente, Genoa, Italy.

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Webb's Academy and Home for Shipbuilders...Sedgwick and 188th streets, New York, N. Y.
Welch, J. J.....The Hollies, Highfield Road, Rock Ferry, Cheshire, England.
Weld, Miss Lydia G.....4801 Huntington avenue, Newport News, Va.
Westinghouse Machine Co , The.....(43284) East Pittsburg, Pa.
Woodwell, Julian E., Inspector, Electric Light Plant, Treasury Department, Washington, D. C.
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EXCHANGES.

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American Marine Engineer, The..... 315 Dearborn street, Chicago, Ill.
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American Society of Civil Engineers.....220 West 57th street, New York, N. Y.
American Society of Mechanical Engineers.....12 West 31st street, New York, N. Y.
Army and Navy Journal.....240 Broadway, New York, N. Y.
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Engineers' Club.....1122 Girard avenue, Philadelphia, Pa.

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Institution of Mechanical Engineers, The
Storey's Gate, St. James Park, Westminster, S. W., London, England.
Institution of Naval Architects.....5 Adelphi Terrace, London, W. C., England.

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Journal of the United States Artillery.....Fort Monroe, Va.

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Northeast Coast Institution of Engineers and Shipbuilders,
4 St. Nicholas' Buildings, West, Newcastle-upon-Tyne, England.

Page's Weekly, Editorial Department,
Clun House, Surry street, Strand, London, W. C., England.
Popular Mechanics.....Journal Building, Chicago, Ill.
"Power,".....World Building, New York, N. Y.

Quarterly Digest of Physical Tests.....1424 N. 9th street, Philadelphia, Pa.
Questions Navales.....200 Rue de Rivoli, Paris, France.

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Schiffbautechnischen Gesellschaft.....Schumanstrasse 2 pr., Berlin, N. W., Germany.
Sibley Journal of Engineering.....Cornell University, Ithaca, N. Y.
Société des Ingenieurs Civils.....19 Rue Blanche, Paris, France.
Steamship, The.....2 Custom House Chambers, Leith, Scotland.
Stevens Indicator.....Stevens Institute, Hoboken, N. J.

Technology Quarterly.....Massachusetts Institute of Technology, Boston, Mass.
Technical World, The Editor.....3321 Armour avenue, Chicago, Illinois.
Tidskrift for Sövesen.....Horten, Norway.

Western Builder, The.....607-8 Montgomery Building, Milwaukee, Wis.
Western Society of Associated Engineers.....Quincy, Ill.
Western Society of Engineers.....1737 Monadnock Block, Chicago, Ills.
Wisconsin Engineer.....Library of the University of Wisconsin, Madison, Wis.

Yacht, Le.....55 Rue de Chateaudun, Paris, France.
Yale Scientific Monthly.....New Haven, Conn.

Zeitschrift Vereines Deutscher Ingenieure.....80 Wilhelmstrasse, Berlin.

ERRATA for JOURNAL OF AMERICAN SOCIETY OF NAVAL ENGINEERS, Volume XVIII, No. 1, *of February, 1906.*—*Paper on "The Bursting of Metal Chambers Under Internal Pressure."*

Plate No. 2: Legend should read "Dispersion of fragments of flask after explosion."

Page 115, first paragraph, second line: For "fitted with a coil and" read "fitted with an afterbody, tail, and"

Page 119, paragraph "Conditions of Test": insert a comma between the words "gun" and "charge" in the first line.

Page 119, paragraph "Conditions of Test," fourth line: For "without the connecting tube" read "without the launching tube."

Page 120, paragraph "Air Flask," fourth line: After the words "cold drawn" insert an asterisk, and at the foot of the page add this foot note: "These flasks are cold drawn from bronze discs, are discs themselves being prepared by hot rolling."